THE MECHANICAL TESTING OF METALS AND ALLOYS

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THE THEORY AND PRACTICE OF STANDARDIZED MECHANICAL TESTING

BY

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THIRD EDITION

LONDON

SIR ISAAC PITMAN & SONS, LTD

1944

First Edition, *1936*
Second Edition, *1938*
Third Edition, *1942*
Reprinted, *1944*

SIR ISAAC PITMAN & SONS, LTD.
PITMAN HOUSE, PARKER STREET, KINGSWAY, LONDON, W.C.2
THE PITMAN PRESS, BATH
PITMAN HOUSE, LITTLE COLLINS STREET, MELBOURNE
UNITEERS BUILDING, RIVER VALLEY ROAD, SINGAPORE
27 BECKETTS BUILDINGS, PRESIDENT STREET, JOHANNESBURG

ASSOCIATED COMPANIES
PITMAN PUBLISHING CORPORATION
2 WEST 45TH STREET, NEW YORK
205 WEST MONROE STREET, CHICAGO
SIR ISAAC PITMAN & SONS (CANADA), LTD.
(INCORPORATING THE COMMERCIAL TEXT BOOK COMPANY)
PITMAN HOUSE, 381–383 CHURCH STREET, TORONTO

BOOK
PRODUCTION
WAR ECONOMY
STANDARD

THE PAPER AND BINDING OF
THIS BOOK CONFORM TO THE
AUTHORIZED ECONOMY STANDARDS

MADE IN GREAT BRITAIN AT THE PITMAN PRESS, BATH
D3—(T.48)

PREFACE

THE demand for another edition of this book has provided an opportunity to enlarge certain sections of the work.

Descriptions of new apparatus have been added, the chapter on Notched-bar Impact Testing has been revised in the light of recent research, and a short account has been given of Damping Capacity in Metals.

Among the several friends from whom I have received assistance, I have especially to thank Mr. Leslie H. Hounsfield, A.R.C.S., M.I.Mech.E., for valuable criticism, and Mr. Chester H. Gibbons, of the Baldwin Locomotive Works, Philadelphia, for kindly supplying particulars of the Southwark-Tate-Emery Machine.

P. F. F.

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1942

PREFACE

TO THE FIRST EDITION

THIS book is the outcome of a series of articles on Testing Machines and their Applications which I contributed to *Machinery* during the years 1931–1932. On considering requests for the publication of the articles in book form, I felt that, while a number of books on the testing of materials were in existence, there was room for one that coupled descriptions of modern testing equipment with its mode of use and which at the same time embraced in a practical way the theory underlying present-day developments in the testing of metals and their alloys. Consequently, the original articles form but a small part of the book. Only such types of testing equipment are described as may be found in up-to-date works, testing rooms, and laboratories. Moreover, some attempt has been made to keep within the range of tests already standardized by the British Standards Institution, or which bear closely on commercial testing.

As the demand on engineering practice becomes more severe, it is reflected in the test room and its personnel. It is hoped, therefore, that the book will be helpful to those whose work brings them into close touch with mechanical testing, and for whom, in fact, the book is mainly intended. Students of Strength of Materials should also find the book of service.

I have adopted the plan of placing references at the end of the book and of indexing them, each with the number of the page to which it refers.

My acknowledgments must be made with respect to sources of information and help. Especially must I thank Professor W. R. D. Jones, D.Sc., for his assistance and criticism throughout the progress of the work. I have also to thank Mr. J. G. Godsell for allowing me to draw upon his extensive experience in matters concerning sheet metals; and Professor W. N. Thomas, M.A., D.Phil.

To the Editor of *Machinery* for permission to make use of the articles contributed to that Journal; to the Institution of Automobile Engineers and The American Society for Testing Materials for allowing me to extract from Papers published in their respective *Proceedings* and which are included among the list of references, I have pleasure in also making acknowledgment.

And in conclusion, I must thank Messrs. Edward G. Herbert, Ltd., Messrs. Alfred J. Amsler, Messrs. Metropolitan-Vickers, Ltd., and other firms who have so generously supplied information, and blocks or photographs for illustrations.

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INSET

THE MECHANICAL TESTING OF METALS AND ALLOYS

CHAPTER I

ELASTICITY : ELEMENTARY THEORY

Introduction. In order to appreciate the results obtained from mechanical tests of materials and to place upon them a proper interpretation, some knowledge of the elementary theory of elasticity is needful. The elastician views materials as of homogeneous structure and the usual engineering calculations are based on this view. Though materials, generally, are far from being homogeneous, the assumption nevertheless serves as a foundation for a vast body of analysis leading to results which the designer can apply with a considerable measure of success. The fundamental parts of the theory as understood by engineers will now be outlined.

Stress, Strain, Modulus of Elasticity. An external force, or load, applied to a body, produces therein an alteration of form. The deformation produced is commonly called the *strain*, while the resistance offered by the molecules of the body in an endeavour to preserve the original form is called *stress*. Quantitatively, *stress* is the load per unit area, tons per square inch or kilogrammes per square centimetre; while *strain* is the fractional alteration in dimensions, length, area or volume, as the case may be—or, in short, the change per unit dimension.

Consider a body in equilibrium under forces F_1, F_2, F_3 and F_4, (Fig. 1). Each small portion of the body will be in equilibrium under certain forces transmitted to its boundary. Imagine the section *ab*, of area A, to divide the body into two parts. The resultant R, of all the forces on one side of the section will be equal and opposite to the resultant of the forces on the other side of the section. Supposing the forces to be normal to and uniformly distributed over the section, the stress there will have the value R/A. If the forces are not uniformly distributed the stress will vary over the section and at any point its value

is defined as the ratio dR/dA, obtained by considering the force acting on the elemental area dA surrounding the point, the area being taken to be vanishingly small while the ratio dR/dA remains finite.

If, after the application and removal of the load, the strain disappears completely, the material is said to be perfectly elastic and the strain is then referred to as *elastic strain.* On the other hand, if the material be permanently deformed by the load, the strain is termed *plastic*, and the body is said to have received a *permanent set.*

When a body is strained by a steady load the strain increases until the stress induced just balances the applied force. A law enunciated by Hooke in 1576, states that, in an elastic body, the strain is proportional to the stress, so that doubling the applied force doubles the distortion produced in the body.

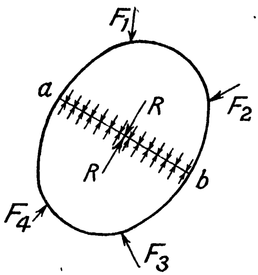


FIG. 1. NORMAL DISTRIBUTION OF STRESS OVER A SECTION

The ratio of the stress to the strain, termed the *modulus of elasticity*, varies for different materials and with the type of stressing applied.

All materials show a limit, the *limit of proportionality*, beyond which stress and strain cease to be proportional. Many materials, in fact, appear to possess no truly elastic range. The limiting value of the stress beyond which a material will fail to recover its original dimensions is termed the *elastic limit.*

Kinds of Stress. The state of stress suffered by a structural member or machine component is, in practice, brought about by a combination of several fundamental types of loading, namely—

(*a*) Tension;
(*b*) Compression;
(*c*) Shear;
(*d*) Torsion; and
(*e*) Flexure or bending. (Fig. 2.)

TENSION AND COMPRESSION. The forces applied constitute a pull or a push and tend to lengthen or shorten the body as the case may be. Extension is accompanied by a lateral contraction while compression is accompanied by lateral expansion.

If l be the initial length of a bar subjected to tension, and l_1 the length after the load has been applied, then $l_1 - l$ is the increase in length. The tensile strain e, or the extension per unit length, is therefore

$$e = (l_1 - l)/l$$

and is a pure ratio. It is assumed that the strain is uniform over the whole length of the rod.

If, for example, $l = 50$ in. and $l_1 = 50{\cdot}1$ in., the strain

$$e = \frac{50{\cdot}1 - 50}{50} = \frac{0{\cdot}1}{50} = 0{\cdot}002$$

In the same way, if d be the original diameter of the bar, supposed round, and d_1 the final diameter, the lateral strain is given by $(d_1 - d)/d$, and in this case is negative.

Similar results apply in compression.

Where tension or compression is concerned the modulus of elasticity is known as *Young's Modulus*; usually denoted by E.

By definition

$$\text{Young's Modulus} = \frac{\text{stress}}{\text{strain}}$$

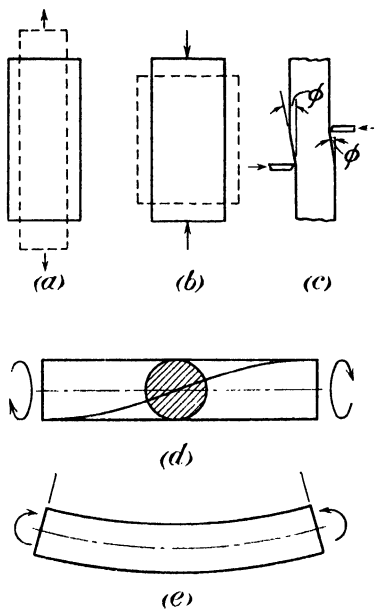


FIG. 2. TYPES OF LOADING

$$= \frac{\text{load/area}}{\text{extension/original length}}$$

$$= Pl/Ae\,.$$

EXAMPLE. A brass wire 0·04 in. diameter and 54 ft. long stretched 0·45 in. under a load of 12 lb.

The sectional area of the wire $= (0{\cdot}04)^2 \times (\pi/4) = 0{\cdot}001256$ in.^{2} The length $= 648$ in.

$$\text{Hence} \qquad E = \frac{12 \times 648}{0{\cdot}001256 \times 0{\cdot}45} = 13\,780\,000 \text{ lb. per in.}^2$$

For bars whose section is other than circular the lateral strain is found by considering the difference between initial and final corresponding dimensions.

The ratio

$$\frac{\text{lateral strain}}{\text{longitudinal strain}} = \sigma$$

is termed *Poisson's Ratio.* It is sometimes defined by its reciprocal $m = 1/\sigma$. For steel m varies from 3 to 4 and in the absence of a more precise figure is frequently taken as 10/3.

SHEAR. The diagram Fig. 2 (*c*) shows equal and opposite forces acting on a bar at some distance apart. Such a system will

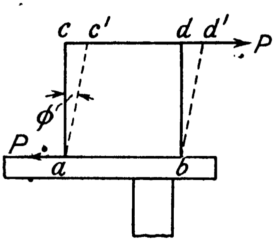


FIG. 3. ILLUSTRATING SHEAR STRAIN

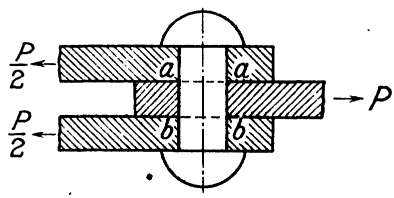


FIG. 4. RIVET IN DOUBLE SHEAR

produce a shear stress in the bar and is convenient for the purpose of illustration, but, in addition to the shear, bending stresses are set up and, in consequence, the member is not subjected to pure shear stress. The production of a pure shear stress in this way is difficult, if not impossible of attainment.

The characteristic feature of a shear is the angular change caused by the application of the load. The phenomenon is well illustrated by considering a block of indiarubber, Fig. 3, one face of which is glued to a table, while to the opposite face is glued a thin strip of wood. If the block of indiarubber is comparatively thin, bending effects may be neglected. The application of a pull P to the strip attached to the top face will result in an equal and opposite pull at the lower face and will distort the vertical faces *ac* and *bd* through an angle ϕ to the positions *ac'* and *bd'* respectively.

The shear strain is measured by the ratio $cc'/ac = \tan \phi$ and as cc' is small compared with ac, $\tan \phi$ may be replaced by ϕ radians. Shear strain is thus measured by the deviation from a right angle.

The ratio (shear stress)/(shear strain) is the *modulus of rigidity*, variously denoted in textbooks by C, G or N.

A practical example of shear stress is the riveted joint, Fig. 4, in which the rivet is in shear at the sections *aa* and *bb*—commonly termed *double shear*.

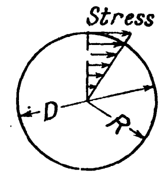


FIG. 5. VARIATION OF SHEAR STRESS IN CROSS-SECTION OF A ROUND SHAFT SUBJECTED TO TORSION

TORSION. Torsion is the shear produced when one layer of a body is made to rotate on the adjacent layer. A cylindrical shaft having equal and opposite torques or couples applied at its ends, and in which the axes of the couples coincide with the axis of the shaft, is subject at every section normal to the axis to pure shear stress. The stress at any point in a section is proportional to the distance of the point from the axis, being zero at the centre of the shaft and greatest at the extreme radius. (Fig. 5.) The external torque T applied, is balanced by the moment of resistance of the section, the relation being

$$T = (2\pi f_s/R) \times (R^4/4) = (\pi/16)D^3 f_s$$

where f_s is the shear stress at the extreme radius R.

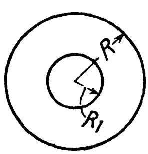


FIG. 6. CROSS-SECTION OF HOLLOW SHAFT

In the case of a hollow shaft of external diameter D and internal diameter D_1, Fig. 6, the relation becomes

$$T = (2\pi f_s/R)(R^4/4 - R_1^4/4)$$
$$= (\pi/16)f_s \, . \, [(D^4 - D_1^4)/D]$$

For the solid shaft $\pi R^4/2$ is the polar moment of inertia J of the section; hence

applied torque $\quad T = f_s(J/R) = f_s Z_p$

where $\quad Z_p = J/R$

is termed the *modulus of the section in torsion*

Imagine one end of the shaft to be fixed whilst the other is

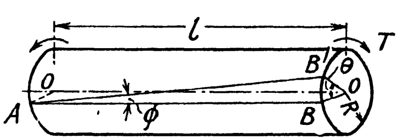


FIG. 7. ROUND SHAFT UNDER TORSIONAL STRAIN

subjected to a couple T, which twists the free end through an angle θ, a radius *OB* moving into the position *OB'*, Fig. 7. A

line AB on the surface originally parallel to the shaft axis becomes distorted to AB'. The angle θ is termed the *angle of twist* of the shaft and from the figure, $AB \times \phi = OB \times \theta$, or $l\phi = R\theta$, the angular distortion being supposed small.

Since shear stress $=$ N $\times$ shear strain, where N is the modulus of rigidity

$$f_s = N\phi = NR\theta/l$$

Consequently, the angle of twist $\theta = f_s l/NR = Tl/JN$ since $f_s = TR/J$.

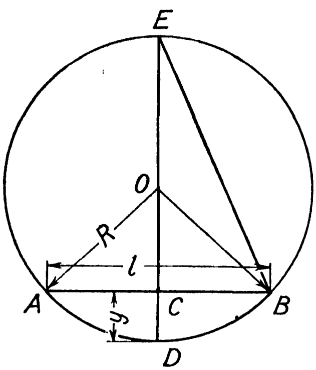


FIG. 8. BEAM BENT TO A CIRCULAR ARC

For a solid circular shaft of diameter D, $J = \pi D^4/32$, and therefore

$$\theta = \frac{32Tl}{\pi D^4 N}$$

If the shaft is hollow, of internal diameter D_1 and external diameter D, then

$$\theta = 32Tl/\pi N(D^4 - D_1{}^4)$$

The product NJ is called the *torsional rigidity* of the shaft.

In the theory of torsion it is assumed that stress is proportional to strain within the elastic limit and that plane sections remain plane after twisting. The circular section is the only one that conforms to the second condition and hence the simple theory of torsion does not apply to sections other than those of circular form. In practical calculations the results of this simple theory are often used in conjunction with a suitable correcting factor.

FLEXURE. The diagram Fig. 2 (e) represents a bar subjected to couples M at its ends, the axes of the couples being at right angles to the longitudinal axis of the bar. The bar or beam originally straight, bends in the plane of the couples to a circular arc, Fig. 8. If R is the radius of curvature of the deflected beam it is easily seen that

$$\frac{DC}{CB} = \frac{CB}{EC} \text{ or } \frac{y}{l/2} = \frac{l/2}{2R - y}$$

that is

$$2Ry - y^2 = l^2/4$$

and as y is comparatively small in any practical case, y^2 may be neglected and the central deflection is given by

$$y \doteqdot l^2/8R$$

It is shown in books on the strength of materials that

$$1/R = M/EI$$

where M is the bending moment, E is Young's modulus and I the moment of inertia of the section of the beam about the neutral axis. It follows that when the bending moment is the same at every section the central deflection of the beam is given by

$$y = Ml^2/8EI$$

In the general case of a beam loaded at right angles to its axis, as for example in the cantilever fixed at one end and loaded at the other, Fig. 9, any section of the beam has to sustain both the bending moment and a direct shear. The bending moment produced by the end load W, at any section distant x from

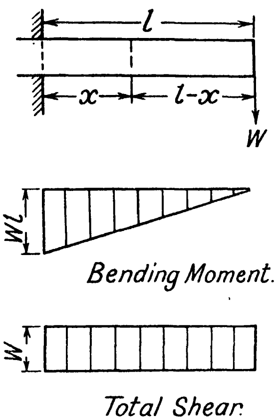
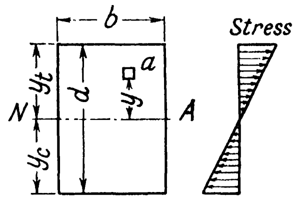


Fig. 9. Cantilever with End Load



(a) (b)

Fig. 10. Flexure

(*a*) Position of neutral axis. (*b*) Variation of stress over the section of a bent beam.

the fixed end is $W(l - x)$ and reaches its greatest value at the point of fixing.

The shear stress is the same at every section and is proportional to W.

The upper fibres of the beam will be in tension and the lower fibres in compression. A particular layer somewhere in the beam will remain unstressed, so far as tension and compression are concerned. The plane of this layer is the *neutral surface* and the trace of this plane on a section perpendicular to the longitudinal axis of the beam, that is, the line *NA*, Fig. 10 (*a*), is called the *neutral axis* of the section.

The stress at any element of area a is proportional to its distance y from the neutral axis and the state of stress over the section is that represented in Fig. 10 (b), the material above NA being in tension and that below NA being in compression.

The neutral axis ordinarily passes through the centroid of the section. This, however, is merely fortuitous and is the outcome of the linearity of the stress distribution over the depth of the section and the fact that the loads act perpendicularly to the length of the beam. For a non-linear stress distribution, or for an initially curved beam, or for a straight beam subjected

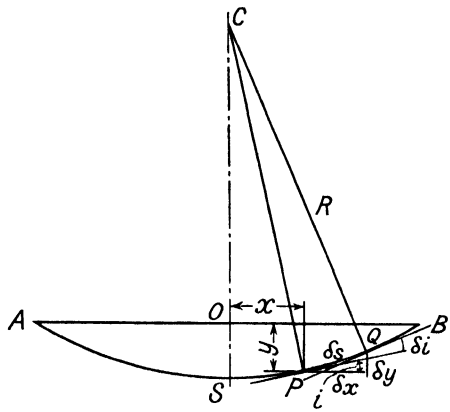


FIG. 11. CHANGE OF SLOPE IN A DEFLECTED BEAM

to oblique loading, the position of the neutral axis is not necessarily coincident with the c.g. of the section.

Textbooks show that if y be the distance of an element from the neutral axis, the stress f there is given by

$$f/y = \mathrm{E}/\mathrm{R} = \mathrm{M}/\mathrm{I}$$

The moment of inertia of the section divided by the distance of the extreme fibre from the neutral axis is termed the *section modulus* Z. The maximum stress occurs at the outermost fibres, and, in general, will have different values in tension and compression.

If f_t and f_c are the respective tensile and compressive stresses at distances y_t and y_c from the neutral surface then, in tension

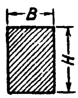
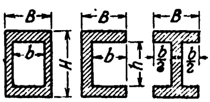
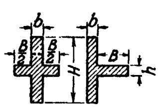
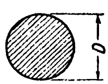
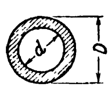
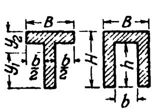
$$\mathrm{M} = f_t\mathrm{I}/y_t = f_t\mathrm{Z}_t$$

and in compression

$$\mathrm{M} = f_c\mathrm{I}/y_c = f_c\mathrm{Z}_c$$

TABLE I

GEOMETRICAL PROPERTIES OF SECTIONS

Section	Area In.^{2}	Moment of Inertia In.^{4}	Modulus of Section In.^{3}
		I	Z
	BH	$\frac{1}{12}BH^3$	$\frac{1}{6}BH^2$
	$BH - bh$	$\frac{1}{12}(BH^3 - bh^3)$	$\frac{BH^3 - bh^3}{6H}$
	$bH + Bh$	$\frac{1}{12}(bH^3 + Bh^3)$	$\frac{bH^3 + Bh^3}{6H}$
	$\pi\frac{D^2}{4}$	$\frac{\pi}{64}D^4$	$\frac{\pi}{32}D^3$
	$\frac{\pi}{4}(D^2 - d^2)$	$\frac{\pi}{64}(D^4 - d^4)$	$\frac{\pi}{32}\left(\frac{D^4 - d^4}{D}\right)$
	$BH - bh$	$I = \frac{(BH^2 - bh^2)^2 - 4BHbh(H - h)^2}{12(BH - bh)}$ $y_1 = \frac{BH^2 - bh^2}{2(BH - bh)}$ $y_2 = \frac{BH^2 - 2bhH + bh^2}{2(BH - bh)}$	$Z_1 = \frac{I}{y_1}$ $Z_2 = \frac{I}{y_2}$

where Z_t and Z_c are the section moduli in tension and compression respectively. Values of I and Z for a few standard sections are given in Table I. When the section is symmetrical about the neutral axis Z_c and Z_t have the same value.

Consider two near points on the neutral surface of a deflected beam at which the radius of curvature is R, Fig. 11. The slope of the beam at P, or its angular deviation from the horizontal there, is i and at Q the slope is $i + di$.

The change of slope from P to Q is thus equal to di, that is to

$$PQ/\mathrm{R} = ds/\mathrm{R} = dx/\mathrm{R}$$

for small deflections.

Further, $i = dy/dx$ and hence the curvature $1/\mathrm{R} = di/dx = (d/dx)(i) = (d/dx)(dy/dx) = (d^2y/dx^2)$. Hence, in terms of M, E and I,

$$d^2y/dx^2 = \mathrm{M/EI}$$

In general, M is a function of the distance x along the beam from some assigned origin and when so expressed in the foregoing equation one integration will enable the slope, and a second integration will enable the deflection to be found for any point along the beam.

The cantilever previously considered provides a simple example.

Here $\mathrm{M} = \mathrm{W}(l - x)$ so that

$$d^2y/dx^2 = \mathrm{W}(l - x)/\mathrm{EI}$$

therefore

$$i = dy/dx = (\mathrm{W/EI})(lx - x^2/2) + \mathrm{C}$$

where C is the constant of integration.

The slope is zero at the fixed end where $x = 0$, hence C is zero.

A second integration gives

$$y = (\mathrm{W/EI})(lx^2/2 - x^3/6) + \mathrm{D}$$

The deflection y is zero at the fixed end and therefore D is zero. The deflection at any point is therefore

$$y = (\mathrm{W/EI})(lx^2/2 - x^3/6)$$

and the maximum deflection δ, which is chiefly what is required, occurs at the free end where $x = l$, so

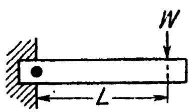
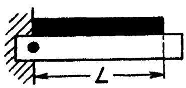
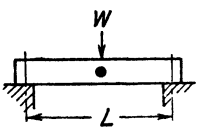
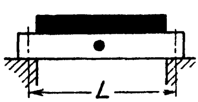
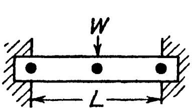
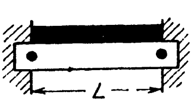
$$\delta = (\mathrm{W/EI})(l^3/2 - l^3/6) = \mathrm{W}l^3/3\mathrm{EI}$$

Some standard cases are given in Table II.

TABLE II

MAXIMUM BENDING MOMENT AND MAXIMUM DEFLECTION OF LOADED BEAMS

W = total load. w = load per inch run.

Type of Loading	Maximum Bending Moment (occurs at ●)	Maximum Deflection
	WL	$\frac{WL^3}{3EI}$
	$\frac{1}{2}WL$ or $\frac{1}{2}wL^2$	$\frac{WL^3}{8EI}$
	$\frac{1}{4}WL$	$\frac{WL^3}{48EI}$
	$\frac{1}{8}WL$ or $\frac{1}{8}wL^2$	$\frac{5}{8}\cdot\frac{WL^3}{48EI}$
	$\frac{1}{8}WL$	$\frac{1}{4}\cdot\frac{WL^3}{48EI}$
	$\frac{1}{12}WL$ or $\frac{1}{12}wL^2$	$\frac{1}{8}\cdot\frac{WL^3}{48EI}$

Bulk Modulus of Elasticity. In addition to Young's modulus (E) and the modulus of rigidity (N), materials possess a volume modulus or *bulk modulus* (K) which represents the ratio between the change of pressure and the change of volume when the material is subjected to a uniform distribution of tension or compression at its outer boundary, as for example a body subjected to hydrostatic pressure.

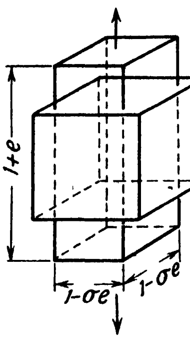


FIG. 12. LATERAL CONTRACTION RESULTING FROM ONE-DIMENSIONAL STRESS

Suppose a cube of unit side to be subjected to uniform tension over one face, Fig. 12. Each stretched edge extends to $1 + e$ and each transverse edge contracts to $1 - \sigma e$. The volume of the cube therefore changes from unity to $(1 + e)(1 - \sigma e)(1 - \sigma e)$.

The change of volume

$$\Delta = (1 - 2\sigma e + e + \sigma^2 e^2 - 2\sigma e^2 + \sigma^2 e^3) - 1$$

which, since the strain is small, is very nearly $e(1 - 2\sigma)$. If each face be subjected to the same pull the total increase in volume will be nearly three times as great or

$$3e(1 - 2\sigma)$$

If p is the applied force we have $e = P/E$ since we are dealing with unit area, and further, on defining Δ as p/K, the relation

$$K = E/3(1 - 2\sigma)$$

A body that possesses the same elastic properties in all directions is termed *isotropic*.

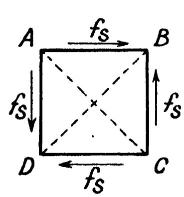


FIG. 13. UNIT CUBE OF MATERIAL UNDER EQUAL SHEARS ON TWO PAIRS OF OPPOSITE FACES

Relations Between the Elastic Constants E, N, K and σ. If shear forces f_s are applied to the opposite faces AB and CD of a unit cube as in Fig. 13, it is clear that for the cube to remain in equilibrium, a balancing couple is required of magnitude $f_s \times AD = f_s$ since AD is unity. This must be accomplished by equal and opposite shear stresses on faces AD and CB as shown in the figure.

For the portion ADC to be in equilibrium a pulling force $2(f_s/\sqrt{2}) = (\sqrt{2})f_s$ must act normal to the diagonal AC and for

the portion BCD to remain in equilibrium a force of amount $f = (\sqrt{2})f_s$ must act towards and normal to the diagonal plane BD. (Fig. 14 (*a*) and (*b*).)

The area of AC or AB is $\sqrt{2}$, as the cube is of unit side, and if f_t and f_c are the respective tensile and compressive stresses $f_t = (\sqrt{2})f_s/\sqrt{2} = f_s$. Similarly $f_c = f_s$.

It follows that shear stresses on planes at right angles to each other are equivalent to tensile and compressive stresses of

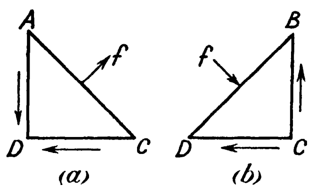


FIG. 14. NORMAL STRESSES ON DIAGONAL PLANES AS A RESULT OF THE STRESS SYSTEM OF FIG. 13

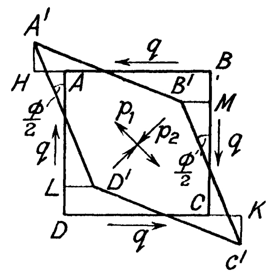


FIG. 15. DISTORTION OF UNIT CUBE BY SHEAR STRESSES

intensity equal to that of the shear stress acting on planes at right angles and inclined at 45° to the planes of shear stress.

Under the action of the shearing forces the cube $ABCD$ of Fig. 15 will be deformed to $A'B'C'D'$, and the shear strain will be given by the angular distortion ϕ.

On planes AC and BD only tensile and compressive forces act, as shown. It is convenient here to adopt a different notation. Let q represent the shear stress on the faces of the cube and p_1 and p_2 the tensile and compressive forces respectively on the diagonal planes. The diagonal AC undergoes a strain p_1/E and also a strain $\sigma p_2/\mathrm{E}$ so that the resulting strain is

$$
\begin{aligned}
p_1/\mathrm{E} + \sigma p_2/\mathrm{E} &= (1/\mathrm{E})(p_1 + \sigma p_2) \\
&= (q/\mathrm{E})(1 + \sigma)
\end{aligned}
$$

as $p_1 = p_2 = q$ in magnitude.

The diagonal BD undergoes a contraction of the same amount.

The strain consists of the horizontal displacements AH and CK on account of the horizontal shearing forces, and the vertical displacements HA' and KC' on account of the vertical shearing

forces. If the total shear is ϕ the angle of 90° between the faces of the cube is changed to $90 \pm \phi$ so that each face turns through $\pm \frac{1}{2}\phi$.

The extension along the diagonal AC

$$\begin{aligned} &= AA' + CC' \\ &= (AH + CK)(1/\sqrt{2}) + (HA' + KC')(1/\sqrt{2}) \\ &= (A'D'/\sqrt{2})(\phi/2) + (A'B'/\sqrt{2})(\phi/2) \\ &= (AD + AB)(\phi/2\sqrt{2}) = 2 \times \phi/2\sqrt{2} = \phi/\sqrt{2} \end{aligned}$$

as the sides of the cube are but slightly altered in length and $AD = AB = 1$.

The strain along the diagonal is therefore

$$(\phi/\sqrt{2}) \div \sqrt{2} = \phi/2.$$

But the strain along the diagonal is, as we have already seen,

$$(1 + \sigma)(q/\mathrm{E}),$$

hence $$(1 + \sigma)(q/\mathrm{E}) = \phi/2 = q/2\mathrm{N}$$

so $$\mathrm{N} = \mathrm{E}/2(1 + \sigma).$$

But $\mathrm{K} = \mathrm{E}/3(1 - 2\sigma)$; hence follow the relations

$$\sigma = \mathrm{E}/2\mathrm{N} - 1 = \tfrac{1}{2} - \mathrm{E}/6\mathrm{K}$$

and $$\mathrm{E} = 9\mathrm{NK}/(3\mathrm{K} + \mathrm{N})$$

connecting the elastic constants.

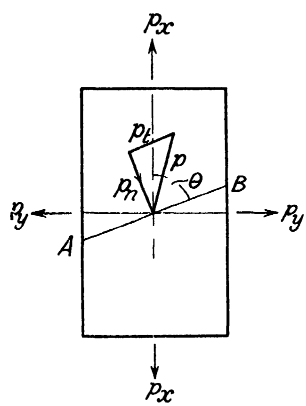


FIG. 16. STRESS COMPONENTS ON INCLINED SECTION OF A BAR SUBJECTED TO STRESSES AT RIGHT ANGLES

Principal Stresses and Planes of Stress. It can be shown that under any complex system of forces the stress at any point in a body can be resolved into a combination of three simple tensile or compressive stresses on three mutually perpendicular planes. Such stresses are called *principal stresses* and the planes are *principal planes* of stress at the point.

A simple case is that of tensile stresses p_x and p_y acting on a block in two directions at right angles, Fig. 16. On any plane AB inclined at θ to the vertical the resultant stress p can be resolved into a stress p_n normal to AB and a stress p_t tangential to AB.

In terms of the applied stresses

$$p_n = p_x \sin^2\theta + p_y \cos^2\theta$$
$$p_t = (p_x - p_y) \sin\theta \cos\theta.$$

and the resultant stress

$$p = \sqrt{(p_n^2 + p_t^2)} = \sqrt{(p_x^2 \sin^2\theta + p_y^2 \cos^2\theta)}$$

The inclination of this stress to the plane AB is given by

$$\tan\alpha = \frac{p_n}{p_t} = \frac{p_x \sin^2\theta + p_y \cos^2\theta}{(p_x - p_y)\sin\theta\cos\theta} = \frac{p_x \tan^2\theta + p_y}{(p_x - p_y)\tan\theta}.$$

If the stresses are unlike, taking p_x as tensile and p_y as compressive, the resulting normal and tangential stresses on the plane AB are

$$p_n = p_x \sin^2\theta - p_y \cos^2\theta$$
$$p_t = (p_x - p_y) \sin\theta \cos\theta.$$

For $p_x = p_y$ and $\theta = 45°$, we have $p_n = 0$ and $p_t = p_x$.

In this instance P_x and P_y are themselves principal stresses.

But consider the wedge ABC of unit thickness Fig. 17, under known stresses p_x, p_y and q as indicated.

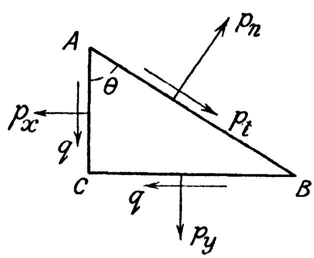


FIG. 17. WEDGE UNDER NORMAL AND SHEAR STRESSES

Resolving normally and tangentially to the plane AB inclined at θ to AC, we find for the normal stress

$$p_n = p_x \cos^2\theta + p_y \sin^2\theta + 2q \sin\theta\cos\theta$$

and for the tangential stress

$$p_t = (p_x - p_y) \sin\theta\cos\theta - q(\cos^2\theta - \sin^2\theta)$$
$$= \tfrac{1}{2}(p_x - p_y) \sin 2\theta - q \cos 2\theta$$

The value of θ that yields the greatest value of p_n is given by $\tan 2\theta = 2q/(p_x - p_y)$, and for the particular value of θ that satisfies this equation p_t is zero.

Further, the equation gives two values of θ differing by a right angle so that there exist two planes on which there is no shear stress. These are the principal planes and the corresponding stresses p_1 and p_2 are the principal stresses. Their values are given by $p_1 = \frac{1}{2}(p_x + p_y) + \frac{1}{2}\sqrt{[(p_x - p_y)^2 + 4q^2]}$

and $p_2 = \frac{1}{2}(p_x + p_y) - \frac{1}{2}\sqrt{[(p_x - p_y)^2 + 4q^2]}$

respectively.

If the x and y directions are to lie in the principal planes the shear stresses must be zero and it can be shown that

$$p_n = \tfrac{1}{2}(p_1 + p_2) + \tfrac{1}{2}(p_1 - p_2)\cos 2\theta$$
$$p_t = \tfrac{1}{2}(p_1 - p_2)\sin 2\theta$$

Equivalent Bending and Twisting Moments. A case of considerable practical importance is that in which one of the normal stresses is zero, the material being under the action of one shear stress and one normal stress, as found in a shaft subjected to combined bending and twisting.

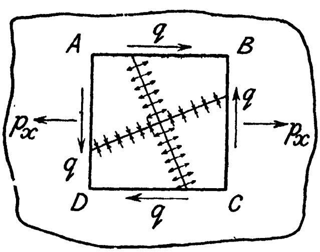


FIG. 18. PRINCIPAL PLANES OF STRESS

Suppose $ABCD$, Fig. 18, to represent a small square prism at the outer surface of the shaft. Under the combined bending and twisting the stresses on the sides of the prism will be the normal stress p_x in the direction of the shaft axis and the shear stress q on the sides of the prism. On certain planes there will be purely normal stress of magnitude obtained by placing $p_y = 0$ in the equations for p_1 and p_2. These equations yield now

$$p_1 = \tfrac{1}{2}[p_x + \sqrt{(p_x^2 + 4q^2)}]$$

and
$$p_2 = \tfrac{1}{2}[p_x - \sqrt{(p_x^2 + 4q^2)}]$$

If M is the bending moment and T the twisting moment applied to a shaft of diameter d,

$$p_x = 32\mathrm{M}/\pi d^3 \text{ and } q = 16\mathrm{T}/\pi d^3$$

Taking the positive sign we have for the greater value of the normal stress

$$p = \tfrac{1}{2}\{32\mathrm{M}/\pi d^3 + \sqrt{[(32\mathrm{M}/\pi d^3)^2 + 4(16\mathrm{T}/\pi d^3)^2]}\}$$
$$= (16/\pi d^3)[\mathrm{M} + \sqrt{(\mathrm{M}^2 + \mathrm{T}^2)}]$$

A twisting moment $T_e = \pi d^3/16$ will produce a pure shear stress and also a pure tensile stress of intensity p, and $T_e = M + \sqrt{(M^2 + T^2)}$ is termed the *equivalent twisting moment.*

It can be shown that the *equivalent bending moment* M_e that would produce the same maximum normal stress as M and T acting together is $M_e = \frac{1}{2}M + \frac{1}{2}\sqrt{(M^2 + T^2)}$.

All planes parallel to the principal planes will be subjected only to normal stresses so that our elementary prism *ABCD* of Fig. 18 will enclose smaller prisms such as *EFGH*, Fig. 19, on whose sides only the normal stresses p_1 and p_2 act. These have the values

$$p_1 = \tfrac{1}{2}p_x + \tfrac{1}{2}\sqrt{(p_x^2 + 4q^2)}$$

and $$p_2 = \tfrac{1}{2}p_x - \tfrac{1}{2}\sqrt{(p_x^2 + 4q^2)}$$

as shown previously.

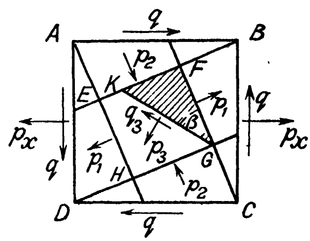


FIG. 19. $\beta = 45°$ IS THE PLANE OF MAXIMUM SHEAR STRESS

Consider now the equilibrium of the wedge of material *FGK* in the prism *EFGH*. On the face *GK* inclined at β to *GF* there exists a normal stress p_3, and a shear stress q_3. The latter is a maximum for $\beta = 45°$ and is then given by

$$q_3 = \tfrac{1}{2}(p_1 - p_2)$$

and in terms of p_x and q, the stresses with which we commenced,

$$q_3 = \tfrac{1}{2}\sqrt{(p_x^2 + 4q^2)}$$

Substituting for p_x and q their values in terms of the bending and twisting moments respectively

$$q_3 = (16/\pi d^3)\sqrt{(M^2 + T^2)}$$

A simple twisting moment $T_e = (\pi/16)d^3 q_3$ would produce the same shear stress. Hence a twisting moment $T_e = \sqrt{(M^2 + T^2)}$ will produce the same maximum shear as M and T acting together.

If the ultimate resistance to rupture by shearing be less than half the resistance to rupture by direct tension or compression a material will fail by shearing when subjected to a direct tensile or compressive force.

The theory that mild steel shafts under combined bending and twisting fail through shear is due to Guest.

In cases of complex stress the plane of maximum stress is not necessarily the plane of maximum strain. This point is of importance in considerations of the criterion of strength; whether failure occurs by the stress exceeding a certain value or by the strain exceeding a certain value.

Suppose p_1 and p_2 to be two principal tensile stresses $p_1 > p_2$. The strain in the direction of p_1 is

$$p_1/E - \sigma p_2/E = (1/E)(p_1 - \sigma p_2)$$

The equivalent stress needed to produce the same strain is

$$p_e = p_1 - \sigma p_2.$$

Hence

$$\begin{aligned} p_e &= p_1 - \sigma p_2 \\ &= (p_x/2)[1 + \sqrt{(1 + 4q^2/p_x^2)}] - (\sigma p_x/2)[1 - \sqrt{(1 + 4q^2/p_x^2)}] \\ &= (p_x/2)[(1 - \sigma) + (1 + \sigma)\sqrt{(1 + 4q^2/p_x^2)}] \end{aligned}$$

At right angles to this there will be an equivalent stress

$$p_e^1 = (p_x/2)[(1 - \sigma) - (1 + \sigma)\sqrt{(1 + 4q^2/p_x^2)}]$$

For steel, Poisson's Ratio lies between 0·25 and 0·3, hence

$$\begin{aligned} p_e &= (p_x/2)[(1 - 0{\cdot}25) + (1 + 0{\cdot}25)\sqrt{(1 + 4q^2/p_x^2)}] \\ &= \tfrac{3}{8}p_x + \tfrac{5}{8}\sqrt{(p_x^2 + 4q^2)} \text{ when } \sigma = 0{\cdot}25 \end{aligned}$$

and

$$p_e = 0{\cdot}35p_x + 0{\cdot}65\sqrt{(p_x^2 + 4q^2)} \text{ when } \sigma = 0{\cdot}3.$$

Graphic Representation of Stress. Referring to the equations for the normal and tangential stress components in terms of the principal stresses, namely,

$$p_n = \tfrac{1}{2}(p_1 + p_2) + \tfrac{1}{2}(p_1 - p_2) \cos 2\theta \qquad (1)$$

$$p_t = \tfrac{1}{2}(p_1 - p_2) \sin 2\theta \qquad (2)$$

these lead to a graphic representation known as *Mohr's Circle of Stress*, Fig. 20.

Let *OA* and *OB* represent the stresses p_1 and p_2 respectively. Then a circle on *BA* of radius $\frac{1}{2}(p_1 - p_2)$ will give the variation in the values of p_n and p_t for various values of θ.

For a given value of θ let CD be drawn making an angle 2θ with the direction of p_n and also the perpendicular DF on OA. From the figure,

$$OF = OC + CF = (p_1 + p_2)/2 + (p_1 - p_2) \cos 2\theta/2$$
$$= p_1 \cos^2 \theta + p_2 \sin^2 \theta$$
$$DF = CD \sin 2\theta = (p_1 - p_2) \sin 2\theta/2$$

so that OF represents the normal stress on the section and DF the tangential stress. The maximum normal stress p_1 is given

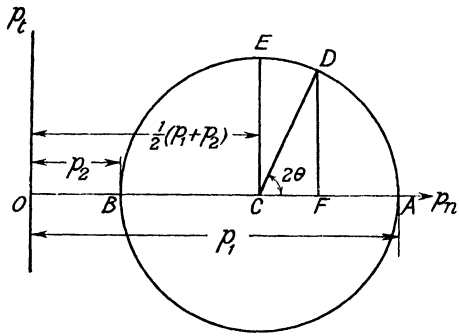


Fig. 20. Mohr's Circle of Stress

by OA and the maximum tangential stress $(p_1 - p_2)/2$ is given by CE and acts on the section corresponding to $\theta = 45°$.

Another useful graphical illustration is the ellipse of stress. Consider a unit cube of material to be subjected to normal stresses p_1 and p_2 as indicated in Fig. 21. Take OX in the direction of p_1 and OY as the direction of p_2. Consider the stresses on the plane AB. Let $OC = f$ be the resultant stress on this plane making an angle ϕ with the vertical. Let CD be drawn parallel to OX and CF parallel to OY. Then $OF = x = f \sin \phi$ and $OD = y = f \cos \phi$. The total force in the direction of OC is $f \,.\, AB$. Its vertical component $f \,.\, AB \,.\, \cos \phi$ balances the vertical force on AB that is $f \,.\, AB \,.\, \cos \phi = p_2 AB \cos \theta$. Its horizontal component $f \,.\, AB \,.\, \sin \phi$ balances the horizontal force on AB; that is

$$f \,.\, AB \,.\, \sin \phi = p_1 \, AB \sin \theta$$

Hence

$$(f^2 \sin^2 \phi)/p_1^2 + (f^2 \cos^2 \phi)/p_2^2 = \sin^2 \theta + \cos^2 \theta = 1$$

Substituting x and y for $f \sin \phi$ and $f \cos \phi$ we have

$$x^2/p_1^2 + y^2/p_2^2 = 1$$

the equation of an ellipse—the ellipse of stress. Its semi-major axis is p_1 and its semi-minor axis p_2 and these are the radii of the outer and inner circles respectively.

To find the stress on any plane such as AB, draw the normal ON. Draw the perpendicular on OY to intersect the ellipse in

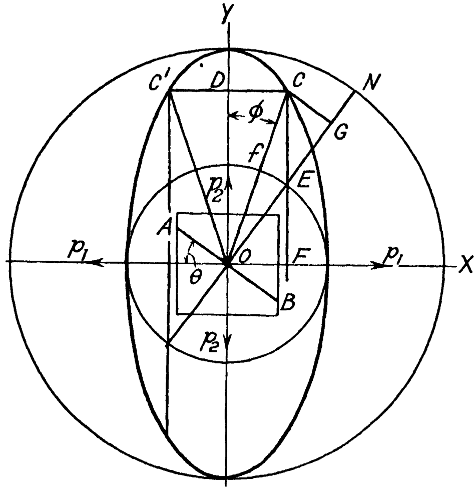


FIG. 21. ELLIPSE OF STRESS

the point C. Then OC gives the magnitude and direction of the resultant stress. A perpendicular CG on the normal will give the normal stress $p_n = OG$ and the tangential stress $p_t = CG$.

If p_1 were negative it would need to be measured in the opposite direction. The resultant stress would then be given by OC'.

Mohr's representation may be extended to three-dimensional systems of stress. Suppose a rectangular bar to be subjected to normal stresses p_1, p_2, p_3 over faces perpendicular to the x-, y-, z- axes respectively, $p_1 > p_2 > p_3$. Over a section through the z- axis the stresses p_n and p_t may be calculated by means of equations 1 and 2, page 18. The circle (1), Fig. 22, represents these stresses. The circle (2) represents the stresses over any section through the x-axis. Circle (3) represents the stresses

on any section through the y-axis. The three circles represent the stresses over three families of sections through the x-, y-, z-axes. For any section inclined to the x-, y- and z- axes the stress components are given by the co-ordinates of a point situated in the shaded portion of the figure. Mohr's Theory, which is an extension of the maximum-shear theory, is dealt

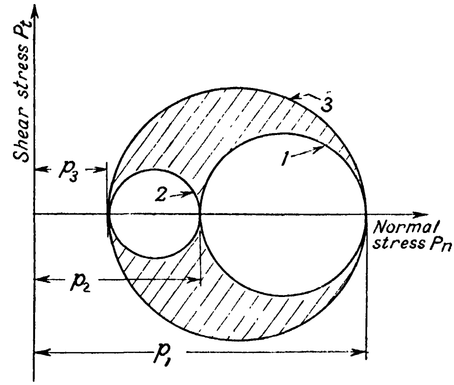


FIG. 22. MOHR'S REPRESENTATION EXTENDED TO SYSTEMS OF THREE DIMENSIONS

with in the works marked with an asterisk in the Bibliography, page 283.

Struts. Cases arise in which a member under load is brought to a state of unstable equilibrium and failure occurs by buckling.

The most important example is that of a strut which, loaded axially, will fail under a load lower than that needed to cause failure through direct stress. The length of the strut and the geometrical distribution of the material in the cross-section are important factors influencing the crippling load.

With struts which are long compared with their lateral dimensions, the collapsing load is given by Euler's formula

$$P = \pi^2 EI/L^2$$

where P = the collapsing load;
E = Young's modulus for the material;
I = the least moment of inertia of the cross-section;
L = the length of the strut supposed pin-jointed or hinged at the ends.

The safe working load is obtained by introducing a suitable factor of safety.

For a given strut the conditions obtaining at the ends greatly affect the collapsing load.

There are four standard cases, Fig. 23—

Case I. Ends pin-jointed or free to take up any angle of slope as in Fig. 23 (*a*). Collapsing load $P = \pi^2 EI/L^2$.

Case II. Both ends fixed in position and direction, Fig. 23 (*b*). Collapsing load $P = 4\pi^2 EI/L^2$.

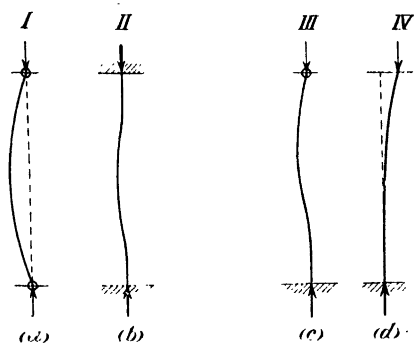


FIG. 23. END CONDITIONS FOR STRUTS—STANDARD CASES

Case III. One end rigidly fixed and the other hinged. Fig. 23 (*c*). Collapsing load $P = (81/4)(EI/L^2)$.

Case IV. One end fixed and the other free to move laterally and to take up any angular position, Fig. 23 (*d*). Collapsing load $P = \pi^2 EI/4L^2$.

The theoretical formulae are derived on the assumption that the struts are perfectly straight and homogeneous and that the load is applied in a perfectly axial manner. Actual struts seldom satisfy the theoretical conditions and various empirical formulae are employed to calculate the collapsing load. Rankine's formula is

$$P = \frac{f_c A}{1 + a(L/K)^2}.$$

where

A = area of cross-section;

f_c = the intensity of the ultimate compressive stress, a

quantity difficult to determine experimentally and sometimes taken as the stress at the yield point in compression;

K = the least radius of gyration of the cross-section;

a = a constant for a strut with both ends free. Case I.

For Case II the constant becomes $a/4$
,, III ,, ,, $a/2$
,, IV ,, ,, $4a$

Values of f_c and a usually given are—

Material	f_c tons per in.^{2}	a
Hard steel . .	30	1/5 000
Mild steel . .	21	1/7 500
Wrought iron . .	16	1/9 000
Cast iron . .	36	1/1 600

Strain Energy. Suppose a structural member or a test piece to be loaded gradually by increasing the load uniformly from zero. If, when the extension amounts to s, Fig. 24, the load has

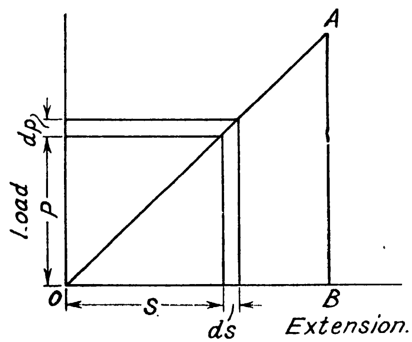


FIG. 24. AREA *AOB* REPRESENTS WORK DONE DURING ELASTIC STRAIN

a value P, then for an extension $s + ds$ the load will be $P + dP$. The average load over the increment of extension ds is $(P + dP/2)ds = Pds$, neglecting the small product $(dP \times ds)/2$. The total work done in straining the member under the assumed conditions is therefore represented by the area OAB and $= \int Pds$.

Now s = strain × length = PL/AE and therefore $ds = (L/AE)dP$. Hence the total work is

$$U = \int \frac{PL}{AE} dP = \frac{P^2L}{2AE}$$

for a member of length L and cross-section A.

The result is obtained on the assumption that the load is applied gradually so that the whole work performed is stored as potential or strain energy in the member. Actually it is immaterial whether the load be applied gradually or not, provided the material is not overstrained in the process, since the excess energy over and above the amount $\frac{1}{2}P \times s$ is transformed into heat during the damping of the vibrations set up.

The strain energy stored in a member due to bending can be shown to be

$$U = \int \frac{M^2}{2EI} dx$$

the bending moment M being variable along the length x of the beam.

Similarly, the strain energy due to shear is

$$U = \int \frac{kF^2}{2AN} dx$$

where A is the area of the cross-section, N the modulus of rigidity, and F the shear force.

The factor k is introduced because the shear stress is not uniformly distributed over the section, the distribution depending on the shape of the section and the loading.

For a circular shaft subjected to a torque T the strain energy is

$$U = \int \frac{T^2}{2NJ} dx \text{ or } \frac{T^2L}{2NJ}$$

if T be constant throughout the length L.

The energy stored per cubic inch of material when stressed to the elastic limit is termed the *resilience*. For a tensile stress p_t, compressive stress p_c or shear stress p_s, the resiliences are $p_t{}^2/2E$, $p_v{}^2/2E$ and $p_s{}^2/2N$ respectively.

If the unit volume of material be subjected to two principal stresses p_1 and p_2 the work done per unit volume may be found as follows—

The strain in the direction of p_1 is

$$e_1 = p_1/E - \sigma p_2/E$$

and in the direction of p_2 is

$$e_2 = p_2/E - \sigma p_1/E.$$

Hence under gradual application of the stress the work done is

$$W = \tfrac{1}{2}p_1e_1 + \tfrac{1}{2}p_2e_2 = p_1^2/2E + p_2^2/2E - \sigma p_1p_2/E$$

Similarly, if at a point three principal stresses p_1, p_2, p_3 act, the work done per unit volume of material in producing these stresses is

$$W = (1/2E)(p_1^2 + p_2^2 + p_3^2) - (\sigma/E)(p_1p_2 + p_2p_3 + p_3p_1)$$

in which a tensile stress is regarded as positive and a compressive stress as negative.

Theories of Elastic Failure. The chief theories put forward to account for the elastic failure of materials from which the strength of a material under combined stress may be deduced from the results of simple tests in tension and compression are—

(*a*) *The Maximum Stress Theory,* sometimes called Rankine's Theory, which assumes that, in ductile materials, yielding starts in an element when the maximum tensile stress becomes equal to the yield point of the material in simple tension, or the maximum compressive stress becomes equal to the yield point of the material in simple compression. This theory is contradicted by many examples.

For instance, if the theory is always true the shearing elastic limit must be at least equal to the tensile elastic limit. But for nearly all metals the elastic limit in shear is much less than the elastic limit in tension.

(*b*) *The Maximum Strain Theory* or St. Venant's Theory, which assumes that the yielding of a ductile material starts when the maximum strain becomes equal to the strain at which yielding occurs in simple tension. Results show that this theory conflicts with practice in many cases.

(*c*) *The Maximum Shear Theory* or Guest's Theory. This assumes that elastic failure begins when the maximum shear stress becomes equal to the maximum shear stress found at the yield point in simple tension. Since the maximum shear stress is equal to half the difference between the maximum and minimum principal stresses the condition for yielding is that

$$\tfrac{1}{2}(p_1 - p_2) = \tfrac{1}{2}\text{ (yield point in tension).}$$

The maximum shear theory agrees better with experiment than either of the foregoing theories, and in machine design is often used in the case of ductile materials. The precise determination of the yield point in shear is not easy.

(*d*) *Haigh's Strain Energy Theory*. This states that inelastic action at any point in a body due to any combination of stresses begins only when the energy per unit volume absorbed at the point is equal to the energy absorbed per unit volume by a bar when stressed to the elastic limit in simple tension.

The condition for yielding is when

$$\frac{1}{2E}(p_1^2 + p_2^2 + p_3^2) - \frac{\sigma}{E}(p_1p_2 + p_2p_3 + p_3p_1)$$
$$= \frac{(\text{stress at yield point})^2}{2E}$$

The above theories are compared graphically in Fig. 25 for the case of two principal stresses. The lines in the diagram represent the values of p_1 and p_2 at which yielding starts according to the several theories. The maximum stress theory is represented by the square *ABCD*. The lengths *OA*, *OB* represent the yield points in simple tension in the x and y directions respectively. Points *C* and *D* correspond to compression.

Point *a* represents equal tensions in two perpendicular directions, each equal to the yield point in simple tension.

By the maximum stress theory there is no yielding for any point inside the square. The maximum strain theory is represented by the rhombus *efgh*. Since a tension in one direction reduces the strain in a perpendicular direction, two equal tensions according to this theory can have much higher values at yielding, represented by the point *e*, than with the maximum stress theory, point *a*. If the two principal stresses are equal and opposite in sign, yielding starts at lower values according to the maximum strain theory—points *f* and *h*—than with the maximum stress theory.

The hexagon *AaBCcD* represents the maximum shear theory. The results given by the maximum shear and maximum stress theories coincide when both principal stresses are equal, but there is considerable difference when the principal stresses are of opposite sign.

In the case of two dimensions we have by the strain energy equation $p_1^2 + p_2^2 - 2\sigma p_1p_2 = (f_y)^2$ where f_y is the stress at the yield point.

This is the equation of an ellipse enclosing all points at which no yielding takes place according to the maximum strain energy theory. It is represented by the full line curve in Fig. 25. As we have seen, the figure enclosing all points at which no yielding takes place according to the maximum shear theory is a

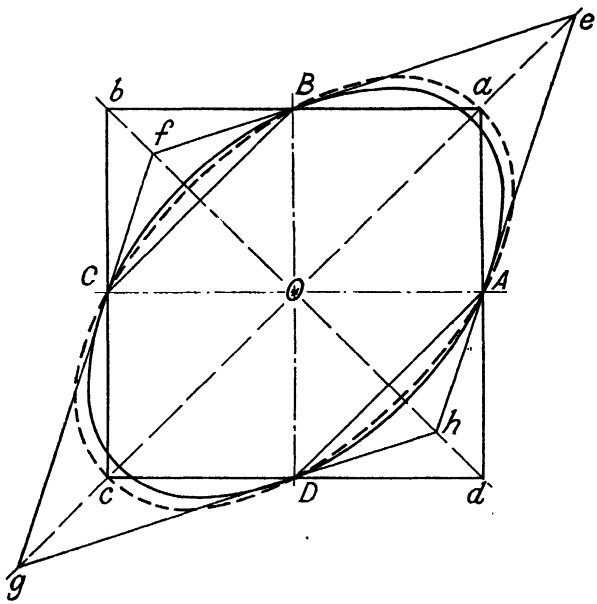


FIG. 25. GRAPHIC COMPARISON OF THE THEORIES OF ELASTIC FAILURE

hexagon. In a three dimensional stress system the corresponding surface is a hexagonal prism; this limiting surface consisting of six different planes in the stress space.

To avoid the discontinuities associated with the maximum shear theory Hencky and von Mises assumed the limiting surface to be capable of representation by an equation of the form

$$(p_1 - p_2)^2 + (p_2 - p_3)^2 + (p_3 - p_1)^2 = \text{constant}$$

the equation of a cylinder circumscribed about the hexagonal prism. It was later shown by Hencky that the expression on the left-hand side of this equation represents, except for a constant factor, the elastic energy stored in the material in shear.

If e_1, e_2, e_3 be taken as the principal strains, the total energy stored in unit volume of material is

$$(p_1e_1 + p_2e_2 + p_3e_3)/2$$

Substituting for the values of the strains in terms of the principal stresses by using the equations (page 12)

$$e_1 = [p_1 - \sigma(p_2 + p_3)]/\text{E etc.}$$

we obtain

$$(1/2\text{E})[p_1^2 + p_2^2 + p_3^2 - 2\sigma(p_1p_2 + p_2p_3 + p_3p_1)]$$

The work absorbed in changing the volume, namely

$$\tfrac{1}{2}[(p_1 + p_2 + p_3)/3](e_1 + e_2 + e_3)$$

is equal to

$$[(1 - 2\sigma)/6\text{E}](p_1 + p_2 + p_3)^2$$

when the foregoing values of e_1, e_2, e_3 are substituted.

Subtracting this work from the total energy stored, there remains for the energy (W) of distortion

$$\begin{aligned} \text{W} &= [(1 + \sigma)/6\text{E}][(p_1 - p_2)^2 + (p_2 - p_3)^2 + (p_3 - p_1)^2] \\ &= (1/12\text{N})[(p_1 - p_2)^2 + (p_2 - p_3)^2 + (p_3 - p_1)^2] \end{aligned}$$

since $\text{N} = \text{E}/2(1 + \sigma)$.

For simple tension where two of the principal stresses are zero

$$\text{W} = (1 + \sigma/3\text{E})p_1^2 = p_1^2/6\text{N}$$

The value of the constant in the equation of the surface is thus

$$12\text{NW} = 12\text{N}(p_1^2/6\text{N}) = 2p_1^2 = 2f_y^2$$

where $f_y = p_1$, is the yield stress in simple tension.

In a two dimensional stress system we have

$$(p_1 - p_2)^2 + (p_2^2) + (-p_1)^2 = 2f_y^2$$

that is
$$p_1^2 + p_2^2 - p_1p_2 = f_y^2$$

the equation of an ellipse. This is shown dotted, circumscribing the hexagon, Fig. 25.

EXAMPLE. To illustrate the foregoing theories consider a shaft of diameter d subjected to a twisting moment T = 25 ton-inches and a bending moment M = 33 ton-inches, the elastic limit of the material being 20 tons per in.^{2} in tension and in shear 0·5 of this value. Suppose the shaft to be strained to half

the elastic limit. The shear stress $q = 16T/\pi d^3$ and the tensile stress $p = 32\ M/\pi d^3$.

Maximum Stress Theory.

The maximum normal stress

$$= \tfrac{1}{2}p + \tfrac{1}{2}\sqrt{(p^2 + 4q^2)}$$

Hence $$\frac{20}{2} = \frac{1}{2}\cdot\frac{32M}{\pi d^3} + \frac{1}{2}\sqrt{\left[\left(\frac{32M}{\pi d^3}\right)^2 + \left(\frac{2 \times 16T}{\pi d^3}\right)^2\right]}$$

Substituting for M and T and solving for d, we find

$$d = 3{\cdot}355 \text{ in.}$$

Maximum Strain Theory, assuming $\sigma = 1/3$,

Maximum stress $= 0{\cdot}35p + 0{\cdot}65\sqrt{(p^2 + 4q^2)}$

and gives $d = 3{\cdot}395$ in.

Maximum Shear Theory.

In this case the allowable limit of stress is reached w shear stress $\frac{1}{2}\sqrt{(p^2 + 4q^2)}$, attains the value $0{\cdot}5 \times$ 2 tons per in.². The required value of $d = 3{\cdot}48$ in.

Strain Energy Theory.

The energy absorbed per in.³ of material is $w = p^2/2E + q^2/2N$. But $N = \frac{2}{5}E$ for steel, hence $w = p^2/2E + 5q^2/4E$.

With the same working stress and $E = 13\ 400$ tons per in.² the permissible amount of energy that may be absorbed per unit volume of material is

$$w_{max} = \frac{1}{2}\frac{f^2}{E} = \frac{1}{2}\frac{(20/2)^2}{13\ 400} = 0{\cdot}00373 \text{ inch-tons}$$

Hence, after substitution,

$$4 \times 0{\cdot}00373 \times 13\ 400 = 2(32M/\pi d^3)^2 + 5(16T/\pi d^3)^2$$

and $$d = 3{\cdot}395 \text{ in.}$$

Using von Mises' Theory

$$p_1 = \frac{1}{2}\left(\frac{32M}{\pi d^3}\right) + \frac{1}{2}\sqrt{\left[\left(\frac{32M}{\pi d^3}\right)^2 + \left(\frac{2 \times 16T}{\pi d^3}\right)^2\right]}$$

$$p_2 = \frac{1}{2}\left(\frac{32M}{\pi d^3}\right) - \frac{1}{2}\sqrt{\left[\left(\frac{32M}{\pi d^3}\right)^2 + \left(\frac{2 \times 16T}{\pi d^3}\right)^2\right]}$$

Substitution in the equation

$$p_1^2 + p_2^2 - p_1 p_2 = f_y^2$$

gives $$100(d^3)^2 = (32^2/\pi^2)(M^2 + \tfrac{3}{4}T^2)$$

whence $$d = 3{\cdot}43 \text{ in.}$$

The relative diameters given by the several theories vary with the conditions specified.

CHAPTER II

THE STRUCTURE OF METALS

THE view of the elastician that materials are isotropic, on which view the analysis of the preceding chapter is based, holds good only with materials such as glass, quickly cooled slags and varnish. These materials are structureless even when viewed under the highest resolving power of the microscope. With metals, however, it has been established beyond doubt that all metallic substances are aggregates of crystals. A crystal may be defined as a substance whose atoms or molecules are arranged in a regular order, following generally some simple geometrical configuration. The regular outward form which we term a crystal is merely a necessary consequence of this internal symmetry. A crystal, therefore, has properties which are not uniform but which vary in different directions. When a member breaks with a "snap" fracture due to shock or fatigue, the crystalline nature is at once apparent and has led to the statement that "because the metal crystallized" it became brittle and broke off "short." This is quite a wrong impression. The material was crystalline from the very commencement of solidification and the crystalline appearance is not the cause, but the outcome, of the mechanism of fracture.

It is agreed that the elastic constants vary in different samples of nominally the same substance and the assumption that a metal or alloy is homogeneous or isotropic is inaccurate. Hence the fundamental assumption of the theory of elasticity is only a rough approximation to the truth. Nevertheless, in any metal or alloy, the crystal grains are generally so very small and their orientation of position, or *lie*, so much at random, that the material may be considered to be statistically homogeneous. This is especially so when we recollect that the sizes of members used in practice are invariably colossal when compared with the dimensions of the crystals of the material. Consequently, for the general purposes of strength of materials the assumption mentioned before may be admitted, and ordinarily, metallic bodies may be considered to be continuous. As soon as precision is attempted the assumption fails completely and in any discussion of phenomena such as elastic failure, strain hardening,

hysteresis, yield, recovery, etc., the fine structure of metals and alloys cannot be left out of consideration. Matter exists in the three states, gaseous, liquid, and solid. The mechanism of transition of matter from one to another of these states is controlled by physical and not by chemical laws. The freezing of steel in an ingot is a process similar precisely to the freezing of water in a pond—both are examples of the change of a physical state from liquid to solid. So also, the condensation of steam in a vessel is the same phenomenon as the condensation of zinc vapour during the manufacture of zinc by the retort process. At the moment of solidification the atoms of the metal arrange themselves in a regular order. During this intense activity of the atoms the rate of cooling is arrested and an amount of energy, as heat, is evolved. This is the well-known *latent heat* of freezing. In the case of most pure commercial metals the atoms arrange themselves in the form of cubes, but zinc, magnesium and antimony crystallize in hexagonal form.

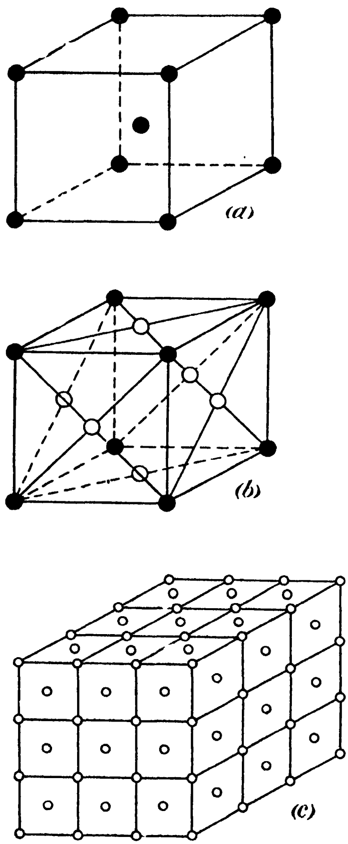


FIG. 26. GRAPHIC REPRESENTATION OF ATOMIC STRUCTURE

The atoms of a metal which crystallizes in the form of cubes, i.e. in the cubic system, may arrange themselves in two ways—(1) They may occupy the corners of a cube with one atom occupying the centre of each cube, Fig. 26 (*a*). This arrangement is called the *body-centred cube* arrangement and it is seen that each crystal unit is composed of nine atoms, kept in their relative positions by forces of attraction and repulsion existing between them. (2) They may occupy the corners of a cube with

one atom occupying the centre of each face, Fig. 26 (*b*). In this case each crystal unit will be composed of fourteen atoms, the arrangement being called the *face-centred-cube* arrangement.

The circles in the figure must not be regarded as representing the actual atoms but merely the spaces within which the atoms vibrate.

Since the atoms are not, as a rule, stationary, but are vibrating, the amplitude of the vibrations increases with the temperature up to a certain point where the vibration of the atoms will overcome the forces keeping them in their respective positions in the cube. At this temperature the regular arrangement is lost and the metal liquefies. Some metals are capable of crystallizing in more than one arrangement and are then said to possess different *allotropic* modifications. Each modification will possess different properties and will be stable at different limits of temperature. Thus, pure iron below 900° C. crystallizes as the body-centred-cube whilst above 900° C. it crystallizes as the face-centred-cube arrangement. Therefore, on cooling, pure iron changes its crystalline form at 900° C. Coincident with these changes is, amongst other things, the loss of the power to dissolve carbon. It is upon this allotropic transformation, which occurs at 900° C., that the whole of the heat treatment of steel depends. There is necessarily a space between the atoms which build up any solid material, so our present idea of a solid is one in which the constituents are built up as a lattice and we thus get the conception of a *space lattice*.

At the moment of solidification the atoms of the substance which have now almost freedom of motion but are under more restraint than when in the gaseous state, are subjected to the crystallizing force which arranges them on a regular space lattice. Some idea of the immensity of this force is shown by the splitting of rocks due to the freezing of water in pores and capillaries. Solidification starts by the simultaneous appearance of nuclei in various parts of the molten metal. Once the nuclei have made their appearance further solidification takes place by the building up of the atoms of the solid in a regular manner around each nucleus. From the nucleus, branches in three directions in space appear. These will have further branches and so on, all in a regular manner, until the solidifying mass of regular space lattice has grown to meet the solid grown in the same manner from another nucleus.

The atoms of the crystallizing material governed by any one

nucleus, will form crystal units which will arrange themselves regularly in one direction, which is generally referred to as the *orientation*. Similarly with the matter crystallizing from other nuclei, the crystal units will arrange themselves regularly in a different direction and this crystallized mass is said to possess a different orientation.

At this stage the solidifying mass is composed of a number of colonies each of which is formed on its nucleus and has been built up of a number of crystal units. The outward shape of the colony is symmetrical, and if the liquid remaining is poured or drains away the solidified mass has the regular shape usually associated with crystals. The perfectly grown crystal is known as an *idiomorphic* crystal. Ordinarily, however, the regular outward shape of a crystal is not seen because the colony, or what is a better term, the *crystal grain*, interferes and is interfered with by its neighbours and the outward form bears no resemblance to a crystal. It is, nevertheless, a crystal in the true sense as it is built up of material in a regular manner and its atoms are positioned on a regular space lattice. This crystal is termed an *allotriomorphic* crystal. The number of nuclei that will appear and hence the number of crystal grains in any particular volume will depend, among other things, on the rate of cooling in the immediate neighbourhood. The greater the rate of cooling the greater the number of crystal grains and generally the tougher the material. The atoms present in the last traces of liquid to solidify will be subjected to the crystallizing or arranging force of contiguous crystal grains and will not be able to attach themselves, as the pull exerted by one grain will balance that of another. The result is that the last remaining atoms will not form part of a crystal but will remain on their irregular or non-crystalline lattice. This film has been given the name *amorphous cement* but a more descriptive term is *irregular lattice*. This boundary between the crystal grains plays an important part. It is harder than the crystal proper, and its atoms not being regularly arranged, there is no crystalline material present and hence no favourable plane of cleavage. Consequently, in metals, fracture should and does ordinarily take place across the cleavage planes of the crystals and not between the crystals, i.e. in the boundaries.

If fracture should take place under normal conditions along the boundaries of the crystal grains a weak or brittle constituent in the boundary material is implied.

Suppose an alloy to contain a small amount of a second metal, and that at some temperature when both are liquid, they are perfectly soluble in each other, so that neither liquid metal can be distinguished from the other, i.e. there is a true homogeneous solution similar to that formed when water is added to alcohol. On solidification one of three things may happen. (1) Neither metal separates from the other and in the solid alloy it is not possible to distinguish either, even under the highest power of the microscope; that is, there is still homogeneity and the conditions of a solution remain, giving what is termed a *solid solution*. The atoms of the second metal take the place of some of the atoms of the parent metal on the space lattice, at first indiscriminately; but later, when the alloy is in a state of equilibrium, the atoms of the added metal are probably more or less uniformly distributed throughout the lattice.

(2) Neither metal is capable of dissolving the other in the solid state and at the point of solidification each constituent metal separates out from the liquid. The effect of adding the second metal to the first is that freezing commences at a lower temperature and that crystals of the metal in excess continue to appear, with the result that the proportion of the second metal in the remaining liquid continuously increases. This, however, does not go on indefinitely until the liquid left consists only of the second metal, because at one particular temperature and composition, i.e. proportion of the second metal, the remaining liquid solidifies at constant temperature by the deposition of alternate crystals (generally thin alternating laminated plates of the two metals).

This composition of the two metals is the mixture which has the lowest freezing point and is called the *eutectic alloy* for the particular series. The solidification of an alloy consisting of an amount of the second metal less than that required for the eutectic alloy, takes place as follows. The metal in excess begins to freeze by the same mechanism as if no other metal were present. After a certain amount of this metal has crystallized the liquid reaches the limiting composition, and in between the crystal grains the alternating laminated structure of the eutectic alloy appears.

The eutectic alloy itself contains excess of neither metal, so no solidification will take place until this lowest freezing point is reached. When this temperature is reached the two metals will

crystallize out simultaneously giving a characteristic laminated pattern. The structure is also grain-like since the eutectic appears as grains or colonies and the laminations of the two constituent metals are parallel in any grain, but the parallel laminations change their direction on going from one grain to another, each eutectic grain having originated in a nucleus and being controlled by it.

If the amount of this second metal is very small, the main or primary crystal will attract its own constituent from the eutectic, and as a result the alloy may consist of crystals of the metal in excess surrounded by an envelope of the second metal. If such a film were composed of a much weaker metal than the parent metal, then the strength of the alloy would be affected adversely and fracture would take place, not, as usual, through the cleavage planes of the crystal but between the crystals. For example, it is known that 0·01 per cent of bismuth will utterly destroy the resistance of copper to deformation. Again, brittleness in mild steel is associated with a film of hard carbide around the iron crystals. A weak film is not always a disadvantage, as a small amount of lead, which will not dissolve in copper or brass, but will separate between the crystal grains, will facilitate rapid machining.

On the other hand, a second metal may act as a stiffener for the rest of the alloy. A slight excess of zinc over that required to form what is known as the α - *solid solution* of zinc in copper causes the appearance of a second solution—the β - *solid solution* —containing more zinc, which being harder acts as a stiffening agent. A large amount of the β - solid makes the brass too hard for cold forging and hot forging must be resorted to.

(3) The two metals, on solidification, may combine in some simple atomic proportion to form a compound which, under the circumstances of the existence of the alloy, is indestructible and plays the part of an element or of a pure substance.

The freezing of this alloy will then take place exactly as in (1) or (2), depending on whether the compound is soluble or insoluble in the metal in excess of that required for the formation of the compound. Intermetallic compounds are generally brittle, and their formation must be controlled in all engineering materials.

It must be pointed out that the foregoing generalizations apply with equal force to alloys consisting of more than two metals, and further, that more than one solid solution and/or

more than one intermetallic compound may be formed in an alloy.

The temperatures at which an alloy undergoes changes such as solidification are generally accompanied by evolutions of heat on cooling and absorptions of heat on heating. If these temperatures are plotted on a chart having the horizontal axis for the composition, and the ordinate as temperature, the resulting chart is called the *equilibrium diagram.* In the case of two metals the equilibrium diagram is a plane figure, but in the case of three the diagram becomes three-dimensional.

The physical changes which a metal or alloy can undergo are not necessarily ended on solidification. Mention has already been made of the allotropic modifications of pure metals, and that on changing from one form to another the properties of the material may, and generally do, change. Another example is the fact that an alloy which at a high temperature forms a solid solution, at some lower temperature may break down into its constituents by a mechanism exactly parallel to the breakdown of a liquid solution into constituents which are not soluble in each other. So complete is the analogy that the eutectic formed from the breakdown of a liquid solution finds its counterpart in the *eutectoid* formed from the breakdown of a solid solution. These changes involve thermal changes which, with modern apparatus, are easily determined. It is upon a knowledge of the changes taking place in the solid alloy that the whole of the heat treatment of alloys depends.

It has already been stated that iron changes its crystal character at 900° C.; above this temperature, iron is in the γ-condition (or face-centred cubic lattice). Below this modification it is called α-*iron* (and is body-centred cubic). Iron when in the γ-condition can dissolve carbon. Consequently, just as magnetic properties are acquired on cooling through 900° C., so is the power of holding carbon in solid solution lost; the carbon, however, is not precipitated as such but as iron carbide, Fe_3C.

It must be remembered, then, that just as the addition of a second metal will lower the temperature at which freezing will start, so will carbon or iron carbide decrease the temperature at which the $\gamma \rightarrow \alpha$ change will take place. However vigorously an iron or steel is quenched the $\gamma \rightarrow \alpha$ change can not be wholly stopped and the speed at which steel cools through this critical range of temperature will affect the structure.

If a steel be heated to about 20° to 30° C. above this critical

temperature for about one hour per inch thickness of section the carbide of iron will go into solid solution due to the $\alpha \rightarrow \gamma$ change taking place.

If the steel be allowed to cool down with the furnace in its own time, the treatment is called *annealing*. If the bar is withdrawn and allowed to cool in air, it is said to be *normalized*, but if the cooling is quickened by plunging the bar straight into a liquid, the bar is said to be *hardened by quenching*—the hardness increasing with the rapidity with which the heat is withdrawn—quenching in cold water giving a much harder product than quenching in oil.

The mechanical properties depend on the type of cooling adopted: annealing gives relatively low tensile figures with relatively high elongations and comparatively low shock-resisting properties, whereas quenching in water gives high tenacity, low elongation, and low resistance to impact if the proportion of carbon be more than 0·25 per cent. Normalizing is roughly intermediate, with good shock-resisting properties.

With relatively slow cooling the nuclei of the α-iron crystals will form in that part of the structure of the γ-solid solution where the atoms are not already on a regular space lattice, i.e. in the grain boundaries, and a structure similar to the γ-iron, but not necessarily of the same grain size, will result if no carbon is present; whereas, if carbon be present the carbide of iron and part of the iron will form a eutectoid, the relative amounts of α-iron and eutectoid depending on the amount of carbon originally in the steel. With a rapid quenching, time is denied for the growth of α-crystals from nuclei and the α-crystals will be precipitated in the weakest part of the iron, i.e. in the cleavage planes. The face-centred crystal unit cubes of γ-iron arrange themselves as octahedra whose section on a cleavage plane is an equilateral triangle. The α-iron is then precipitated in these cleavage planes and can be seen under the microscope with suitable magnification, like crystal plates arranged at 60° to one another. This structure is typical and denotes a very brittle material. It is not confined to iron-carbon alloys (steels), but is found in any alloy system which crystallizes in a similar way, such as aluminium-copper-alloys (Al-Bronze) containing more than 10 per cent of aluminium.

The preparation and examination of metals and alloys under

the microscope requires a technique which is attained only by practice. A suitable specimen is carefully cut from the bulk and ground down to a mirror-like finish by means of emery papers of increasing degrees of fineness, the last traces of fine scratches being removed by polishing the specimen on a rapidly rotating cloth-covered disc fed with some oxide abrasive, generally diamantine, which is calcined alumina Al_2O_3. This fine polishing causes a very thin surface layer of "flowed" metal which has, during its slight mobility, smeared the surface and filled the little valleys of the scratches.

The structure of the metal is then revealed by an etching reagent which un-builds the structure downwards by dissolving away, first of all, the smeared surface layer. Dilute alcoholic solutions of mineral acids, such as nitric acid, are suitable reagents. The etching is stopped when desired and the structure examined under a microscope by normal illumination.

We have seen now that in any metal the atoms are built up in a regular manner having a geometrical configuration, and we can visualize that certain planes will contain more atoms than others and will accordingly resist stresses better so that cleavage and fracture will take place on certain planes more readily than on others. It will be realized that the lattice of any metal can be distorted, but only within certain limits, and that once that limit is exceeded the lattice breaks down.

On the basis that the atoms of any two metals are different, it follows that, when any atoms of an added metal take the place of atoms on a regular lattice there is bound to be distortion and consequent hardening. Such is the case when zinc is added to copper. Copper crystallizes in the face-centred cubic arrangement. A certain number of zinc atoms can enter the lattice, and although distorting the lattice and making the material harder it is not until about 30 per cent of zinc is added that the lattice is distorted so much that it breaks down.

Brass containing up to 30 per cent zinc is a solid solution termed α, and the space lattice pattern of the crystal is still the face-centred cubic. This brass can be cold forged. When more than 30 per cent zinc is added the number of zinc atoms so distorts the lattice that it breaks down and a new lattice, the body centred cubic, is formed, which is not so ductile. This second space lattice is another solid solution called β.

Ordinary yellow brass or Muntz Metal contains 40 per cent zinc and consists of crystals of the α-solid solution surrounded

by the crystals of the β-solid solution. This amount of β-solid solution is sufficient to stiffen up the α-solid solution so that it can be forged only at high temperatures, the alloy being too brittle to be forged at ordinary temperatures.

The space lattice of a metal or alloy can also be deformed by mechanical means. When the lattice is only slightly distorted, on removing the stress the lattice reverts to its normal configuration. If, however, the limit of distortion is exceeded, the lattice is permanently deformed and the elastic limit has been passed and a permanent set appears. Above the elastic limit, i.e. in the plastic region, the crystals accommodate themselves by a process whereby the planes of atoms slip over one another. This can be seen on a previously polished and etched surface of a metal or ductile alloy by the appearance of parallel, more or less straight, lines known as *slip bands* which, in reality, are little steps on the surface which have been produced by minute slips occurring on some of the crystal planes. By this means the crystal grains are elongated, becoming fibre-like, so that when a stress is imposed gradually and continuously up to breaking point, the resulting fracture is fibrous, breaking on a cleavage plane of the long fibre.

If, however, a stress greater than the breaking load is placed suddenly on the specimen the crystal grain has no time to elongate, and fracture occurs across the cleavage plane of the non-elongated crystal grain and appears coarsely crystalline.

In this short summary dealing with the structure of metals sufficient has been said to enable the testing engineer to appreciate the importance of the inner structure of the materials with which he has to deal. For a more detailed account reference should be made to works on metallography.

CHAPTER III

UNIVERSAL TESTING MACHINES

Principles Underlying the Construction of Testing Machines. The essential parts of a machine for making tension and compression tests comprise a straining gear for applying the load to the test piece and apparatus for measuring the load, so arranged that the accuracy of the measuring device is unaffected by the distortion of the specimen.

The straining mechanism may consist of a hydraulic ram on which fluid pressure is exerted by means of an accumulator or by a pump; or the straining action may be accomplished by screw gearing, operated by hand or power.

Machines may be of the horizontal or vertical type, the former offering some advantage for heavy work although occupying more floor space.

For measuring the load or force on the specimen several methods are available. The chief of these are—

(1) An application of the principle of the steelyard employing a lever, or system of levers, provided with a movable counterpoise. Such machines are termed *single-lever* or *multiple-lever* machines, as the case may be.

(2) The use of a weighted pendulum, the load being determined by observing the angle through which the pendulum is deflected.

(3) The simple method of measuring the pressure of the fluid in the straining cylinder.

(4) Balancing the load by fluid pressure acting on a diaphragm.

While methods (3) and (4) largely obviate inertia stresses in the test piece they are not regarded with favour, and by responsible authorities machines embodying the weighbeam principle are preferred*

Lack of space prevents the description of the many types that have been introduced and only a few modern machines used for commercial testing and research will be considered.

Denison 15-ton Testing Machine. A typical single-lever

* Accurate and reliable Bourdon gauges for use with testing machines have been developed by the Baldwin-Southwark Corporation, Philadelphia.

machine of moderate capacity is the 15-ton machine made by Messrs. Samuel Denison & Son Ltd., Leeds.

The principle of the machine is shown in Fig. 27. A vertical frame F supports the horizontal steelyard or beam L, at the knife-edge B, the angular motion of the beam being limited by stops SS. At C is a second knife-edge carrying the link D, to the lower end of which the specimen is attached. The other end of the specimen is gripped in the cross-head H connected by tie rods to the ram R working in the straining cylinder A. The downward movement of the ram under pressure loads the test piece. To balance the load, the poise w is moved along the steelyard until equilibrium is attained.

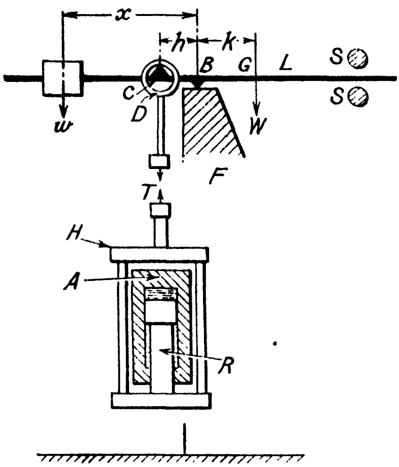


FIG. 27. PRINCIPLE OF SINGLE-LEVER TESTING MACHINE

If W is the weight of the steelyard acting at its centre of gravity G, at a distance k from the support, w the weight of the poise and T the pull on the test piece acting at a distance h from the support; then, for equilibrium,

$$\mathrm{T}h + wx - \mathrm{W}k = 0$$

The zero position, x_0 of the poise weight is obtained by placing $\mathrm{T} = 0$ in the above equation, whence

$$x_0 = \mathrm{W}k/w$$

The actual machine is illustrated in Fig. 28. For tensile tests the machine is equipped with dies for holding unprepared rounds, flats, and squares. The grips will accommodate flat specimens up to $\frac{3}{8}$ in. thick, and round and square specimens from $\frac{1}{4}$ in. to $1\frac{1}{4}$ in. diameter or side.

The maximum and minimum distances between the wedge boxes P are 18 in. and zero respectively.

Compression tests can be made on specimens up to 8 in. in length and the compression plates A are 7 in. diameter. The

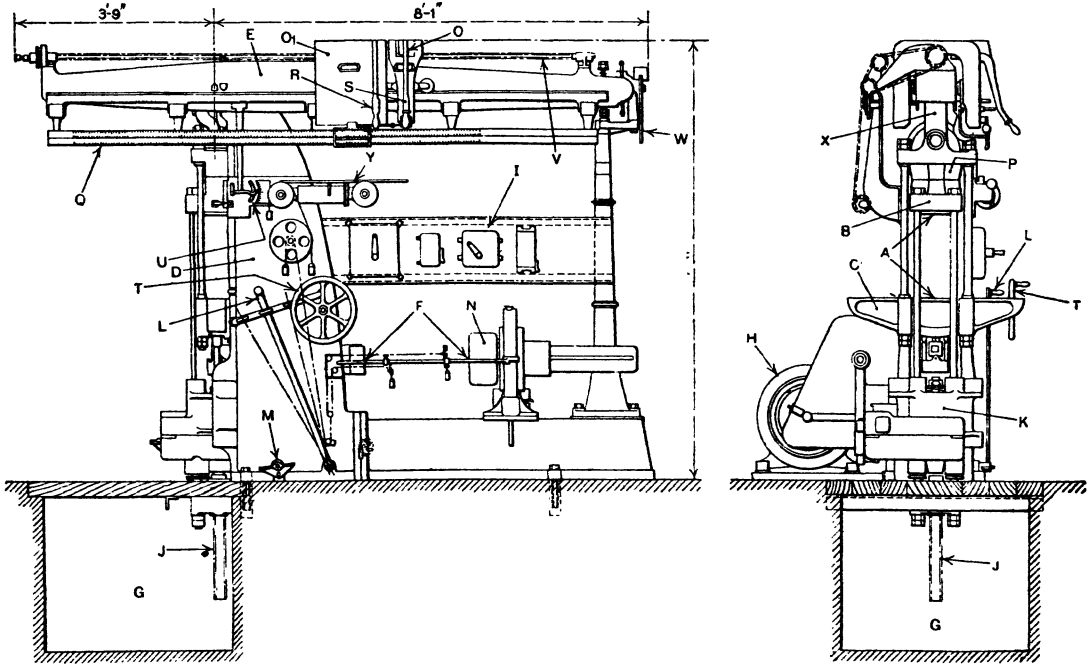


FIG. 28. DENISON 15-TON VERTICAL SINGLE-LEVER TESTING MACHINE
(*Machinery*)

upper compression plate is attached to the underside of the straining head *B* by a simple bayonet fastening and the lower plate is located on the beam *C* by a spigot. The faces of the compression plates are scored with concentric rings to facilitate setting the specimen.

Transverse tests can be made on spans from 6 in. to 36 in. by 6 in. increments. Tests in double shear are performed by using the upper compression plate to apply the load to the middle member of a shear dog. The dog is provided with holes for testing specimens $\frac{1}{4}$ in., $\frac{3}{8}$ in., and $\frac{1}{2}$ in. diameter.

Bending tests on bars to be bent through 180° are carried out by the use of a special knee and block.

The controls are conveniently placed on the front of the main column *D*. Torsion tests are made at *F*, the maximum and minimum observed lengths of the torsion specimens being 20 and 3 in. respectively.

The straining gear is on the left of the main standard. No special foundations are required with the exception of a small pit *G* to take the straining crosshead.

The drive is by electric motor *H* through a roller chain and the motor is provided with a speed control panel *I*, the range of speed being from 350 to 1 400 r.p.m.

The straining gear receives its motion from one of the two chain drives from the motor to the lay-shaft and thence through a worm gear to the straining screw *J*. The two chain drives from the motor in conjunction with friction clutches controlled by the lever *L* provide two speeds, the upper being used for setting the head and the lower for applying the load.

The setting of a dog clutch controlled by a pedal *M* determines whether the lay-shaft drives the torsion straining head or the tension straining gear. For torsion at *F*, the drive passes through a compound worm box to the torque sleeve *N*, with which is incorporated a floating transmitter to allow for variation in the length of the torsion specimen when under test.

The weighing gear for use in tests other than torsion comprises a steelyard with a travelling poise weight OO_1. The knife-edges are of a patent dove-tailed inclined type, and as they are jig-produced they are interchangeable. The poise OO_1 is in two parts, a traveller and a follower, the former being $\frac{2}{15}$ of the total weight. With the traveller and follower disconnected the capacity of the machine is reduced to two tons. A hand lever *R* and catch on the follower enables the two to be readily

connected and the machine brought back to its full capacity. *S* is the lever for disengaging the poise nut and *T* the poise drive handwheel.

The weight of the poise leader is 100 lb. and the weight of the leader and follower together 750 lb.

The graduations on the scalebar are—

(1) 15 tons by $\frac{1}{10}$ ton.
(2) 2 tons by $\frac{1}{100}$ ton.
(3) 15 000 lb.-in. by 100 lb.-in.

In the foregoing the vernier on the poise weight reads down to

(1) $\frac{1}{100}$ ton.
(2) $\frac{1}{1000}$ ton.
(3) 10 lb.-in.

On front of the steelyard within the operator's range of vision when observing the specimen, is a small bell-cranklever—the steelyard balance indicator *U*—having its long arm horizontal when the steelyard is at balance and oscillating up and down in unison with the steelyard. The pointer of this bell-crank lever travels over a scale with three sets of graduations: (1) from − 0·5 ton to + 0·5 ton by 0·05 ton; (2) from − 0·5 ton to + 0·5 ton by 0·005 ton; (3) from − 400 lb.-in. to + 400 lb.-in. by 50 lb.-in. The resistance to motion of the steelyard is by two pairs of carefully calibrated springs. Should the steelyard be out of balance the operator is quickly aware of it and sees the error of the steelyard reading within the plus or minus readings of the scale.

The two pairs of springs are for use with the combined poise weights, i.e. 15 tons capacity. A lever *W* is provided so that one pair may be disengaged when testing with the small poise weight only, i.e. 2 tons capacity. In this case the 0·05 ton scale would be used on the bell-crank indicator. If by mischance the specimen breaks with the steelyard out of balance, the error as shown by the graduated scale needs to be added to or subtracted from the steelyard reading in order to obtain the correct value.

An autographic stress-strain recorder *Y* is fitted which works in conjunction with the poise weight.

The overall dimensions of the machine are: length 11 ft. 10 in.; width 5 ft. 1 in.; height 7 ft. 11 in. The approximate weight is $3\frac{1}{2}$ tons.

The Buckton 100-ton Horizontal Testing Machine is shown diagrammatically in Fig. 29, from which the principle of operation will be readily understood. In the straining cylinder C works a single-acting ram carrying the slide S. The slide is guided by a groove in the bed. The crosshead H_1 carried by the slide can be locked in any one of a number of positions by means of keys.

Tension specimens are held between the crossheads H_1 and H_2, the latter being attached to stout steel rods D running the

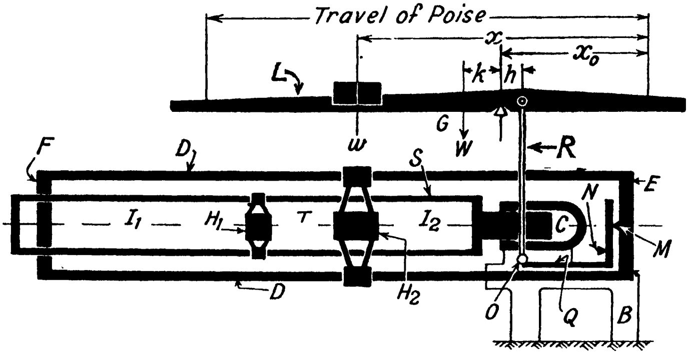


FIG. 29. BUCKTON 100-TON HORIZONTAL TESTING MACHINE

whole length of the machine. The rods are connected to the crossheads E and F, one at each end of the machine, the whole being termed the *measuring frame.*

The action of the machine will be understood by imagining a specimen inserted between H_1 and H_2 and pressure applied to the straining cylinder. As the ram moves outwards carrying with it the slide and its attached crosshead, the pull on the specimen is transmitted through the head H_2 and the rods D to the knife-edge M in the head E. The motion of E is constrained by the bell-crank lever Q and a system of knife-edges. The pull on the specimen is ultimately transmitted through the rod R to the steelyard L. By adjusting the poise to balance, the load on the test piece can be ascertained.

Compression and bending tests can be made by placing the specimen in either of the positions I_1, I_2. Bending tests are generally made by using a substantial cross beam as a support. This is attached to the head F and will accommodate specimens up to 10 ft. in length.

The friction between the sliding frame and the bed is of little import as the force to overcome frictional effects here is not transmitted to the measuring gear. Movements of the measuring frame, though of course very small, should not be restricted by friction, and to secure this condition as far as possible the frame is supported on a system of knife-edges and rollers.

In testing long members in compression, any side thrust caused by eccentricity of loading on the machine, or bending of the specimen, is transmitted by the slide to the bed without inconvenience. Some extra provision, however, is needed in the case of the measuring frame and this is secured by means of side stays fitted with rollers.

Pressure is applied to the straining cylinder by a hydraulic accumulator operating at 2 000 lb. per in.^{2} The return stroke of the ram was brought about in the earlier machines by the action of a heavy counterweight working in a pit under the bed, but this is now replaced by a hydraulic return cylinder.

The outer end of the ram at its point of attachment to the crosshead is threaded and carries a large circular nut. With this device it is possible to lock the specimen and maintain on it a constant load for as long as desired. Any trouble that might be occasioned by release of part of the load through fluid leakage is thus avoided.

The knife-edges M and N are carried by the bell-crank lever Q, their seats being formed by extensions on the head E and the cylinder C respectively. When the machine is operated at its full capacity the load on the knife-edges is 5 tons per in. run.

The joint O and the knife-edge N are on the same level and 32 in. apart. The vertical distance between M and N is 4 in., thus giving a leverage of 8 to 1.

A duplicate pair of knife-edges, not shown in the diagram, is provided at a distance of 8 in. below the level of the knife-edge M. This permits the leverage to be altered from 8 to 1 to 4 to 1 with a corresponding alteration of the scale of the weighbeam from 100 tons to 50 tons.

The poise w weighs $\frac{1}{2}$ ton and on the 100 ton scale the poise travels 2 in. per ton of load. With the 4 to 1 leverage the poise travels twice this amount or 4 in. per ton of load. On this scale it is possible to read to 0·001 ton, roughly $2\frac{1}{4}$ lb.

It is of interest to consider in some detail the forces acting on the steelyard. Referring to Fig. 29, if T_0 is the pull in the rod R due to the weight of the parts, W the weight of the beam

acting at its centre of gravity G, and w the weight of the poise acting at a distance x_0 from the fulcrum, then for equilibrium:

$$Wk = T_0h + wx_0$$

For a load P on the specimen producing a pull T in the rod R, we have, with the poise on the other side of the fulcrum and distant x from the zero of the scale,

$$Wk = (T + T_0)h + w(x_0 - x)$$
$$= Th + T_0h + wx_0 - wx$$

and since $$T_0h = Wk - wx_0$$
$$Wk = Th + Wk - wx_0 + wx_0 - wx$$

so that $$Th = wx.$$

The distance h is 8 in. so that if the machine is used on the 100 ton range with the 8 : 1 leverage and if there is a pull of P tons on the specimen, $T = P/8$.

Hence $(P/8) \times 8 = wx = \frac{1}{2}x$, giving $P = x/2$ tons.

If P is 1 ton, $x = 2$ in., or the scale is 2 in. per ton of load, and 200 in. of travel are needed for the full 100 tons.

With the poise at the extreme end of its travel

$$8\ (T + T_0) = Wk + w(200 - x_0)$$
$$= Wk + 134w$$

as x_0 is about 66 in., T is then 100/8 and we have

$$8\ T_0 + \frac{100 \times 8}{8} = Wk + w(200 - x_0)$$

that is $$100 = Wk + w(200 - x_0) - 8T_0$$
$$= Wk + 134\,w - 8T_0$$

Neglecting then the weight of the parts, we see that the force at the specimen when the poise is at the end of its travel is $134w$, and since w is $\frac{1}{2}$ ton the poise in this position balances about $\frac{3}{4}$ of the total load, the remainder being balanced by the weight of the beam itself.

Some interesting data relating to inertia effects in a machine of this type were given some years ago by the late Professor A. C. Elliott of University College, Cardiff.

The weight of the beam was about 1·25 tons and its moment of inertia about the fulcrum approximately 49 ton.-ft.^{2} units. The moment of inertia of the poise when at the end of its travel

about the fulcrum was 63 ton-ft.^{2} units. The total moment of inertia I of the beam and poise was thus 112 ton-ft.^{2}

Now, assuming the test piece to be stretching under an acceleration α, the angular acceleration of the steelyard will be $8\alpha/h$ and the torque about the fulcrum $(8\alpha/h) \times \mathrm{I}$.

The corresponding force at the specimen would be

$$\frac{8\mathrm{I}\alpha}{h} \times \frac{8}{h} = 112\alpha \times \frac{8^2}{(8/12)^2} = 16\ 100\alpha,$$

the reduced mass of the steelyard and poise being represented by 16 100 tons at the specimen.

The periodic time of vibration of the beam is given by

$$\mathrm{T} = 2\pi\sqrt{\frac{\text{angular displacement}}{\text{angular acceleration}}}$$

and this is the same as the period of oscillation of the specimen about its mean length, namely

$$\mathrm{T} = 2\pi\sqrt{\frac{\text{displacement}}{\dfrac{\text{force on specimen}}{\text{reduced mass}}}}$$

If L is the length of the specimen, A its cross-sectional area, E the modulus of elasticity, S the reduced mass, e the displacement, and $g = 32{\cdot}2 \times 12$ in the present instance,

$$\begin{aligned}\text{Force on specimen} &= \text{stress} \times \text{sectional area}\\ &= \mathrm{E} \times \text{strain} \times \text{sectional area}\\ &= \mathrm{E} \times (e/\mathrm{L}) \times \mathrm{A}\end{aligned}$$

and the time of vibration

$$\mathrm{T} = 2\pi\sqrt{\frac{\mathrm{LS}}{g\mathrm{EA}}}$$

For a wrought iron bar $1\frac{1}{2}$ in. diameter and 36 in. long, for which E was 12 000 tons per in.^{2}, and under a load of 20 tons for which the reduced mass of the beam was about 7 300 tons, the calculated time of oscillation was 1·12 sec. while experiment showed the time to be 1·5 sec.

An experiment on a mild steel test piece 10 in. long between gauge points showed an amplitude of vibration of 0·0006 in., corresponding to the full swing of the beam. On the whole length of specimen the amplitude was about 0·0007 in. The big difference between the observed and calculated results is

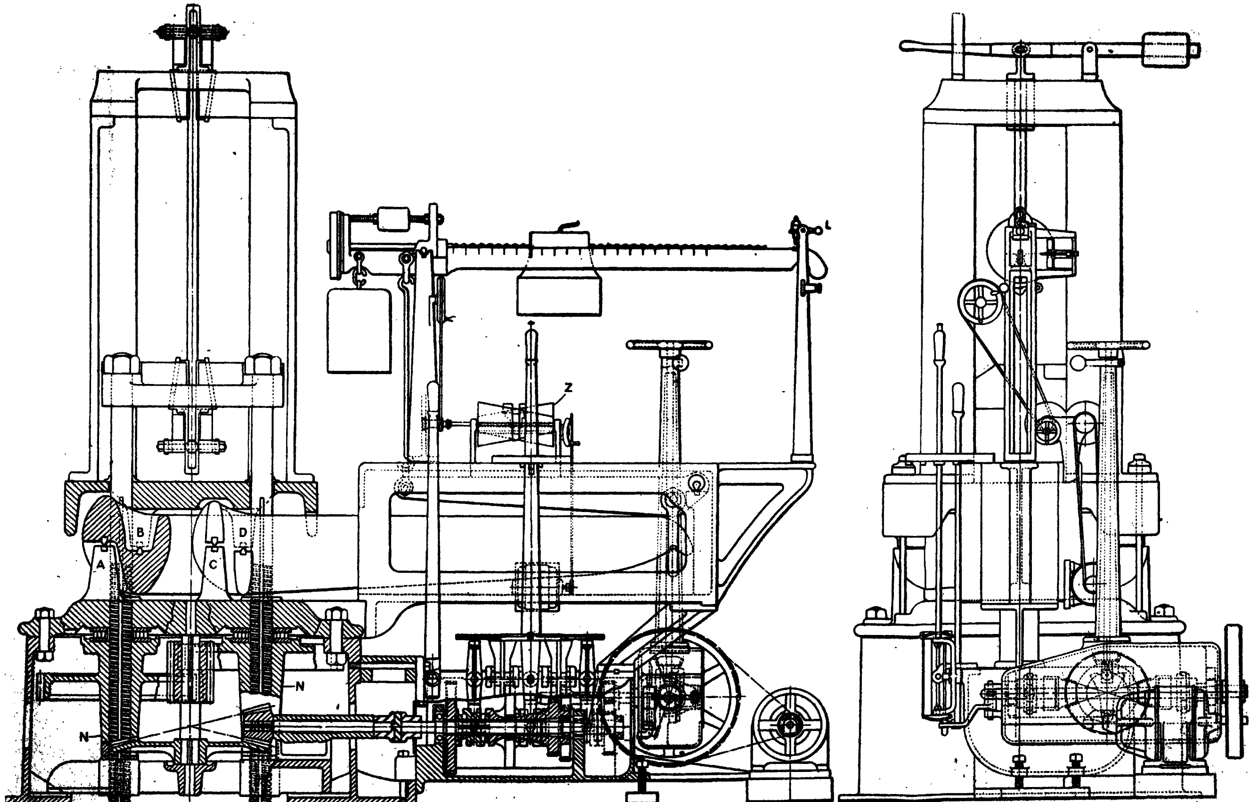


FIG. 31. ELEVATION AND SECTIONAL VIEWS OF OLSEN MULTIPLE-LEVER TESTING MACHINE
(*Machinery*)

due to lost motion in the parts and will be smaller the greater the leverage of the machine.

The manufacture of the Buckton machines is now in the hands of Messrs. W. & T. Avery Ltd., Birmingham.

Lever System of Multiple-lever Machine. The lever system of a multiple-lever machine is shown diagrammatically in Fig. 30. The straining screws pass through bearings in the base *F* and

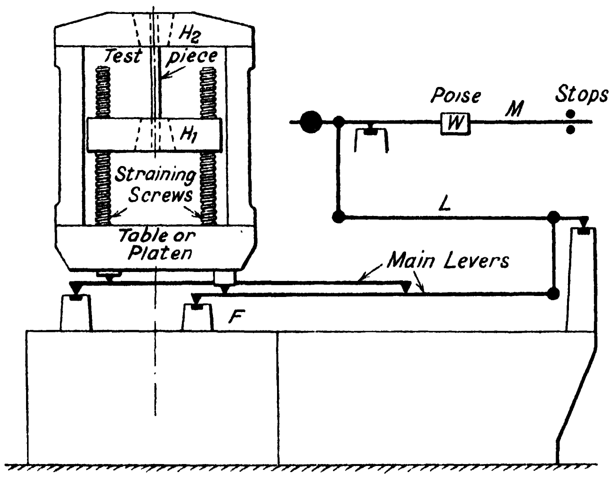


FIG. 30. LEVER SYSTEM OF MULTIPLE-LEVER MACHINE

rotate in bronze nuts in the pulling head H_1. The downward pull on the specimen is transmitted through the head H_2 to the platen resting on the main levers, and is then transmitted through the intermediate lever *L* to the steel yard *M*. Owing to the great leverage ratio, about 3 000 to 1 at the end of the steel-yard, the poise need weigh only a few pounds.

The testing speeds available vary with the size of the machine. In the case of a machine giving six pulling speeds they are approximately—

8·0 in. per min. for setting the head.
2·0 in. per min. for quick testing.
1·0 in. per min. for medium testing.
0·4 in. per min. for slow testing.
0·2 in. per min. for crushing tests.
0·1 in. per min. for slowest speed.

The Olsen Multiple Lever Machine is shown in Figs. 31 and 32. In this machine the straining screws work in rotating nuts

N and do not project above the pulling head. The space around the tension specimen is thus less restricted than would otherwise be the case, which is sometimes an advantage when an extensometer is used. A pit underneath the machine is needed to accommodate the straining screws. Two, three, or

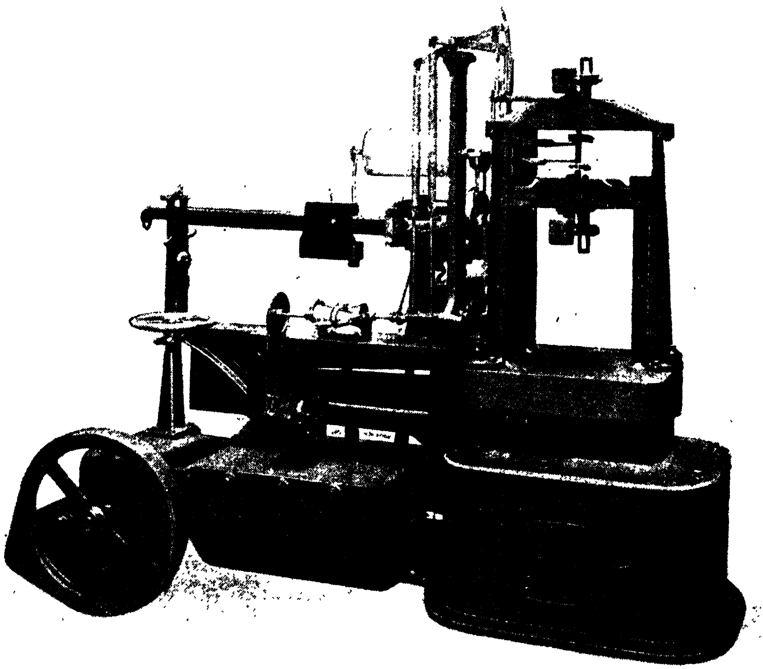


FIG. 32. REAR VIEW OF OLSEN MULTIPLE-LEVER TESTING MACHINE
(*Tinius Olsen Testing Machine Co.*)

four screws may be employed and the nuts are of exceptional length in order to minimize wear.

The main levers rest on fulcra as indicated at A and C, Fig. 31, and support the platen on knife-edges as at B and D. The recoil of the system when the specimen fractures is taken by rubber buffers on the top corners of the platen. Three poise weights are provided, one for registering full load, one for half load, and one giving $\frac{1}{10}$ full load. An adjustable counterweight

enables the weight of the specimen and grips to be balanced before commencing a test.

An interesting feature is the autographic gear for recording the test and its automatic control. The autographic gear is shown in Fig. 33 and is entirely independent of the weighing system.

The clamps *B* are attached in position on the specimen by means of a jig or setting device. The calliper fingers *A* rest in contact with the clamps on the specimen. The elongation of the specimen is transmitted through the calliper fingers to the tubes *C* and *D* respectively. Relative movement between the tubes is then converted into rotation of the drum *R* by means of the thin metal band *L* fastened at *K*. This rotation provides the strain ordinate.

The screw on the steelyard controls the motion of the poise *V* through variable speed cones *Z* (Fig. 31) which are driven by an independent electric motor. The poise *V* moves the pen carriage *T* on its guide *Q* a corresponding amount by means of the cord *W* passing under pulley *VA* and over pulley *VB*. The combined motion of the pencil and drum develops the load-extension diagram.

In one type of recording gear the pencil is kept in a state of vibration by an electromagnet and traces out the record as a succession of dots the object being to avoid pencil friction. The speed cones permit the rate of travel of the weighing poise to be easily and quickly adjusted in order to produce a characteristic diagram under all conditions.

The method of using the autographic attachment is as follows.

The beam is balanced with the poise at zero and the clamps attached to the specimen as in Fig. 33. The hand lever *O* is latched down at *I*, which is the locking position, when the calliper fingers may be adjusted on their respective tubes to the required positions. The calliper fingers should be so placed as to be from half an inch clear to lightly touching both sides of each clamp *B*. In the tension test it is necessary that the upper calliper fingers be placed on the vertical tube *C* in front of the machine, while the lower calliper fingers are attached to the tube *D* at the rear.

The two tension springs *G* and *H* at the rear of the machine should be so placed that the heavier of the two springs will be at the position shown at *G*. The hand lever *O* is then unlatched from position *I* and raised to the extreme position where it

is latched at *J*, and at the same time the calliper fingers are guided to their respective clamps to ensure proper seating.

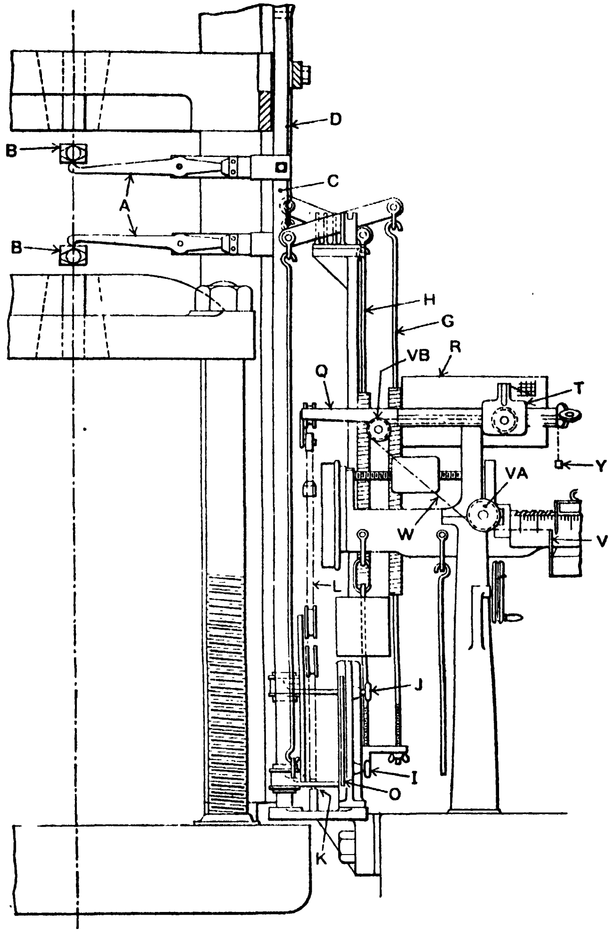


FIG. 33. AUTOGRAPHIC ATTACHMENT FOR OLSEN TESTING MACHINE
(*Machinery*)

The upper calliper fingers drop on to the lower clamp. If this action is not secured the spring tension on *G* and *H* should be adjusted.

The chart is placed on the drum and the pen set to zero. The motion of the pen on its guide Q is obtained by fastening the cord w to the poise. The cord is kept taut by the counterweight Y.

A switch is then closed to complete the circuit through the top contact of the beam stand at the extreme right of the machine. The small lever L, Fig. 31, is raised and latched in its upper or operating position to diminish the motion of the beam. The load is then applied at a suitable rate and the speed of the poise adjusted by means of the speed cones to give vibration of the beam during the test. When the ultimate load is reached the switch is opened and a second switch closed in order to complete the circuit through the lower contact on the beam stand. This reverses the motion of the weighing poise and permits a record of decreasing load to be obtained.

The Olsen machine is now made in this country by Messrs. Edward G. Herbert Ltd., Manchester.

The Amsler Testing Machine. A machine in which no steelyard is employed is made by Messrs. Alfred J. Amsler & Co., of Schaffhouse, Switzerland. It is operated by oil pressure and consists of three parts: the testing machine proper, an oil pump for producing the pressure and a dynamometer for measuring the load applied to the test piece.

The machine proper is composed of a cast iron base supporting two or more vertical steel columns. The upper ends of the columns carry a crosshead to which the straining cylinder is attached. When the ram is forced upwards by the oil pressure it lifts a crosspiece which carries a cradle suspended by steel rods. The cradle carries the upper grips for the tensile test as in Fig. 34; the lower grips are attached to the base of the machine. On the upper face of the cradle provision is made for the equipment for compression, bending, and shear tests.

No collars or cup leathers are used in the straining cylinder to prevent leakage. Instead, the clearance between the ram and the cylinder wall is kept so small that a very slight leakage of oil takes place past the ram. By this means friction is eliminated and the total load on the test piece may be measured with considerable exactness from the pressure of the oil alone.

The pumps are specially designed to provide uniform delivery of oil to the straining cylinder without pulsation.

The straining cylinder is connected with the dynamometer through the pipe A which ends at the valve B. From B a pipe

leads to a small cylinder containing a piston *D*. This communication is always open and cannot be cut off by the valve *B*, so that the dynamometer is always loaded to the same pressure as exists in the straining cylinder. The outward movement of the piston causes the pendulum *P* to deflect and thus to

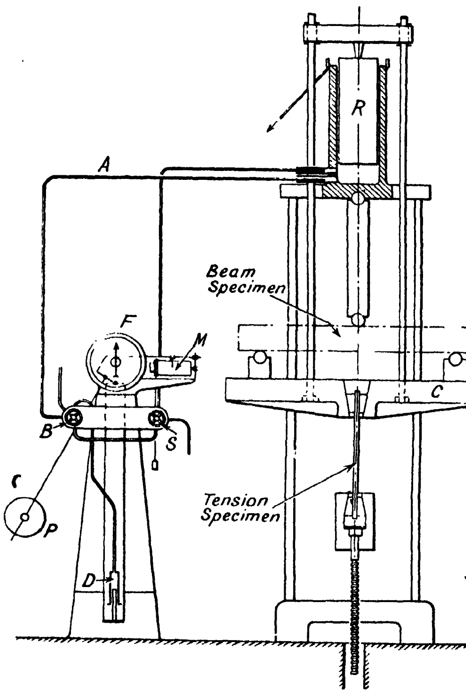


FIG. 34. DIAGRAMMATIC VIEW OF AMSLER TESTING MACHINE

indicate on the dial *F* the load exerted on the test piece. The drum *M* enables autographic records to be obtained.

The makers guarantee these machines to 1 per cent absolute accuracy for all loads above half the indicated maximum. All machines comply with the regulations laid down by the International Association for Testing Materials.

The Avery Self-indicating Machine. The Universal Self-indicating Machine recently introduced by Messrs. W. & T. Avery, Ltd., is illustrated in Figs. 35 and 36. The straining

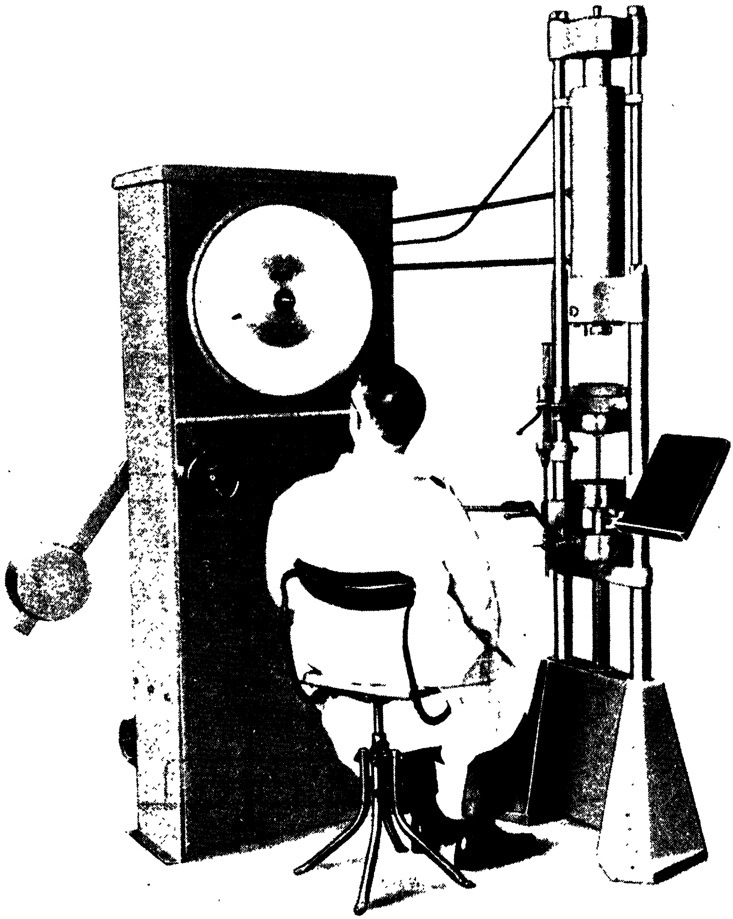


FIG. 35. AVERY SELF-INDICATING TESTING MACHINE

of the specimen and the weighing of the load are effected through the medium of hydraulic pressure supplied simultaneously to the straining cylinder and to a small cylinder housed in the indicating cabinet. A motor-driven gear pump provides the means of obtaining the necessary pressure.

The straining unit consists of a single cylinder placed at the

top of the machine, the ram operating without packings. The rate of straining is controlled through a graduated wheel marked "strain setting control" in Fig. 36. By moving the straining or load-holding lever to an intermediate position a given load can be sustained on the specimen.

Four ranges of capacity, in the ratios 10, 5, 2, and 1, are

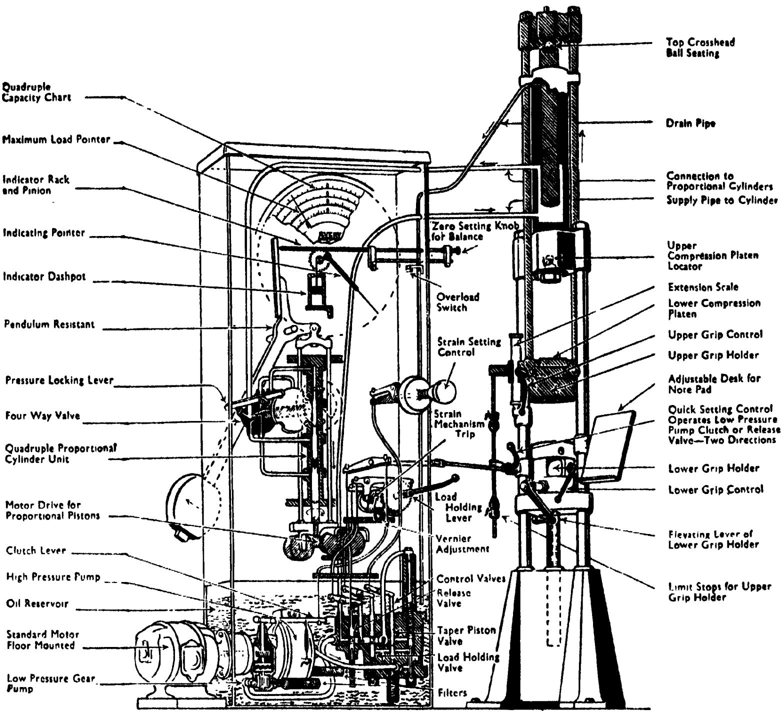


FIG. 36. SECTIONAL VIEW OF AVERY SELF-INDICATING MACHINE

provided on the chart of the load indicator. A heavy pendulum provides the necessary resistant to the pressure in the indicating cylinder. The use of several charts avoids the need for changing the pendulum weight when it is desired to change the capacity. The whole arrangement is clearly shown in Fig. 36, which is self-explanatory.

These machines have been designed specially for convenience of manipulation, and are made in standard capacities of 10, 30 and 50 tons.

The Southwark-Tate-Emery Machine. The Southwark-Tate-Emery Universal Testing Machine, manufactured by the Baldwin Southwark Division, Baldwin Locomotive Works, Philadelphia, is a development of the large precision-testing machine built in 1879 by A. H. Emery for the Watertown Arsenal, U.S.A.

The straining action is produced by fluid pressure acting on a ram in the usual way, and weighing is effected by means of a hydraulic capsule through which the load must of necessity pass. Connected to the capsule is a smaller hydraulic cylinder and ram loaded through a weighing scale by dead weights.

This machine is unique in being the first to employ the auxiliary hydraulic cylinder in association with the capsule, and to employ flexure plates in the lever system of the weighing mechanism and in the testing machine itself. The flexure plates take the place of the knife edges commonly employed.

The balance or scale for weighing the load has a very high leverage ratio, the combination of this with the hydraulic magnifier giving a ratio of 420 000 to 1. The balance is an elaborate affair of extreme accuracy and sensitivity enclosed in a glass-fronted case. The late Professor W. Cawthorne Unwin, well known for his wide experience in the testing of materials, was eulogistic in his praise of the Emery machine.

In the machine as now developed for commercial purposes the weighing scale is replaced by a Bourdon tube and a system of springs. The arrangement of a vertical machine of this type will be clear from the diagram (Fig. 37A).

Upward movement of the ram causes the load, either tension or compression, to be imposed on the sensitive platen arranged between the ram table and the upper crosshead. The straining cylinder is cast integral with the base of the machine and its head forms part of the hydraulic capsule which is essentially a rigid cylinder and piston unit having 0·1 inch clearance in the bore. The piston is maintained central and free from contact with the casing by a saw steel bridge ring bearing on a flexure ring on the piston at its inner edge and on a second concentric flexure plate on the casing ring at its outer edge.

A diaphragm, the purpose of which is solely to prevent oil leakage, covers the end of the piston and is carried on the bridge ring across a relatively wide gap between the piston and the casing ring. This gap is purposely enlarged so as to reduce to a minimum the bending stress in the diaphragm,

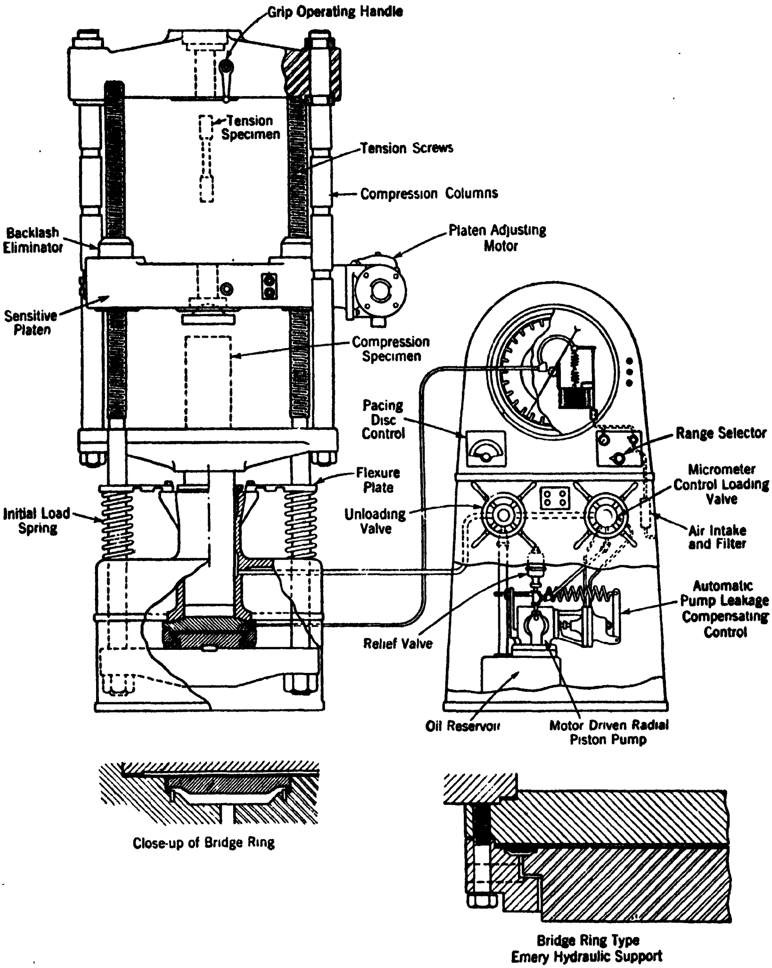


FIG. 37A. ARRANGEMENT OF THE SOUTHWARK-TATE-EMERY UNIVERSAL TESTING MACHINE

where it is clamped around its periphery. The fluid in the capsule is a light oil whose viscosity changes but little with temperature. The maximum thickness of this film is about 0·03 inch, and since there is no motion of the surfaces on each other all loads are transmitted to the fluid in the weighing

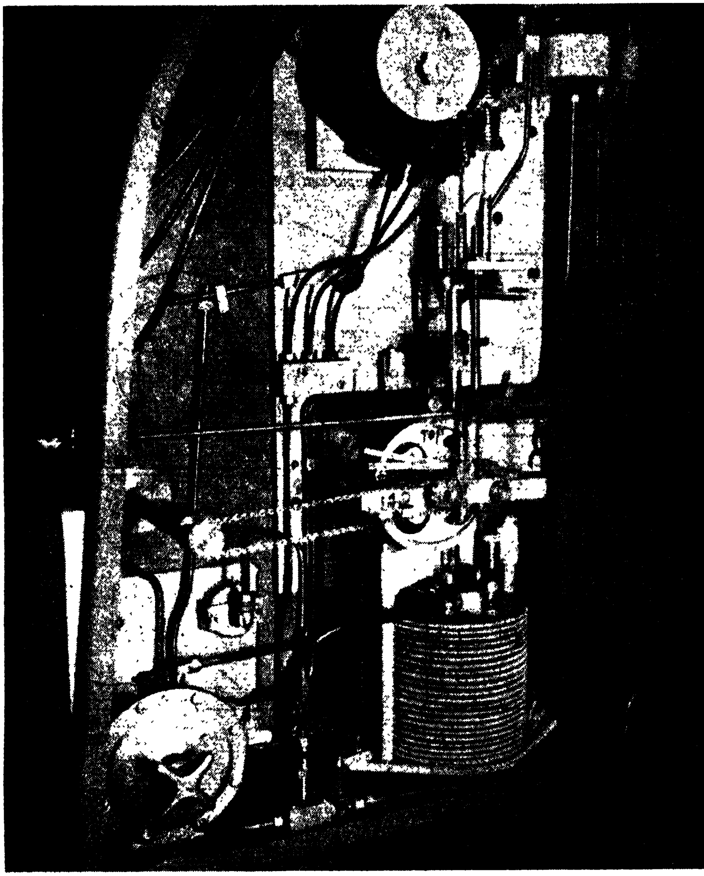


FIG. 37B. SOUTHWARK-TATE-EMERY MACHINE: INDICATOR MECHANISM

system without loss. Pressure is carried to the Bourdon tube through a thick-walled copper pipe. Great care is taken in the manufacture of the Bourdon tubes themselves and they are specially treated and aged before use.

Initial load springs support the dead weight of the weighing frame and maintain a datum pressure on the capsule. Effects of eccentric loading are taken care of by the combined action of the bridge ring in the capsule and the two flexure plates at the upper end of the straining cylinder. The straining pressure

is supplied by a motor-driven radial-piston pump. The indication of the load is accomplished by means of the Tate-Emery Null Method Indicator. A servo-motor, an outside source, is arranged to nullify movement of the end of the gauge element due to changes in the hydraulic pressure by applying, through a system of springs, a restoring force to the free end of the Bourdon tube. The load so applied is indicated on a 24-inch dial giving a scale length of 66 inches. The operation is performed by air which enters under pressure through a reducing valve.

FIG. 37C. 4,000,000-LB. SOUTHWARK-EMERY TESTING MACHINE: UNIVERSITY OF CALIFORNIA

The springs are of Iso-Elastic metal, an alloy of the Elinvar type. The materials commonly used for springs are unsuitable for precise work owing to their deviation from proportionality, modulus changes, and hysteresis and creep effects. The arrangement of the indicator mechanism and the Bourdon tubes with their attached springs are shown in Fig. 37B. The indicator may be graduated to read to 0·5 lb. in the case of a machine of 120 000-lb. capacity. The guaranteed inaccuracy is less than one-half of one per cent, or 0·1 per cent of the range capacity, whichever is the greater. The loading range may be changed during a test and without interrupting it, by the turn of a knob.

The external power source provides all the power needed to

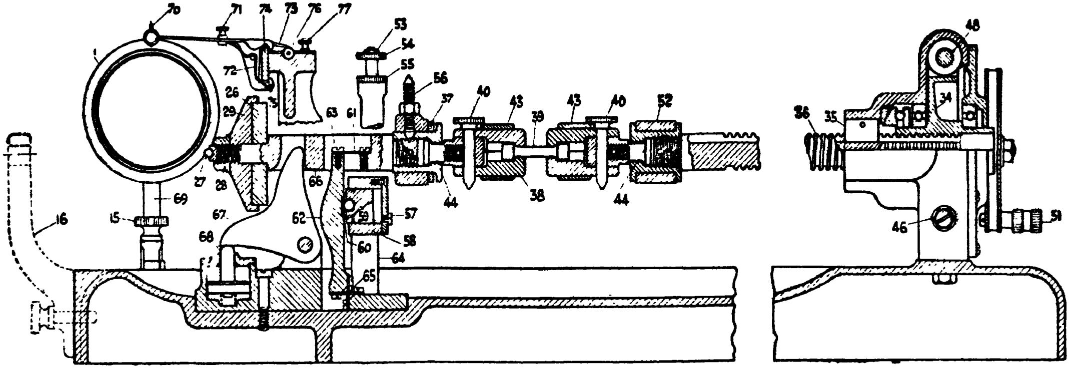
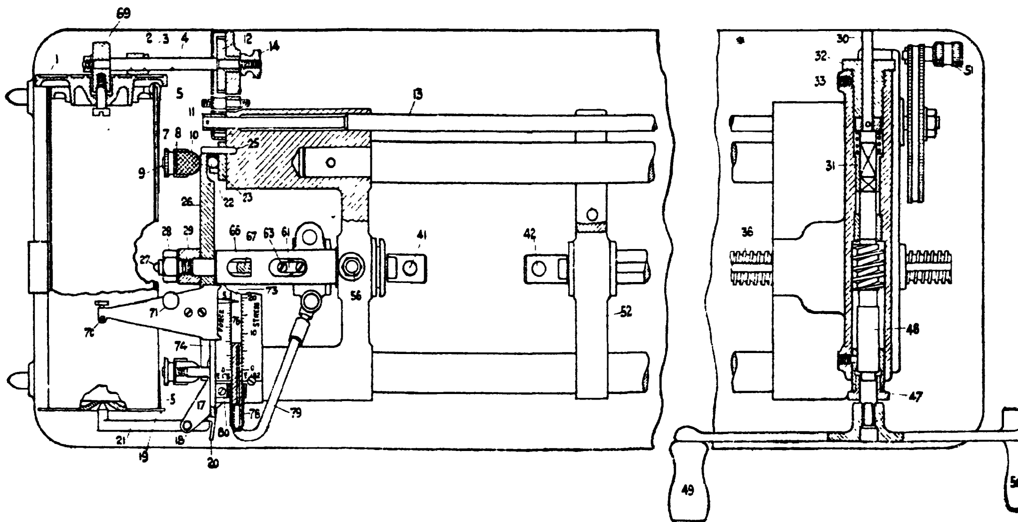


FIG. 38. HOUNSFIELD TENSOMETER

1. Recorder drive plate
2. Recorder sliding driver
3. Locating device for 2
4. Recorder driving spindle (short)
5. Graph retaining spring
7. Nuts for rubber buffer
8. Rubber buffer housing
9. Stud for 8
10. Rubber buffer
11. Recorder driving pinion
12. Recorder driver gear wheel
13. Long recorder driving shaft
14. Nut for spindle 4
15. Nut for recorder bracket
16. Brinell attachment bracket
29. Bridge piece
30. Spindle for motor drive
31. Square coupling clutch
32. Bearing for 30
33. Lock screw for 32
34. Worm wheel
35. Square key for operating screw
36. Operating screw
37. Rubber ring
38. Chuck
39. Test piece
40. Pin for attaching chucks
41. Nose piece (left)
42. Nose piece (right)
43. Chuck ring
56. Stop screw
57. Mercury drain screw
58. Mercury cylinder
59. Mercury piston
60. Space to left of piston
61. Coupling blade spring
62. Lever
63. Screw for 61
64. Distance or calibrating piece
65. Fulcrum hinge
66. Tension bar
67. Shock bell crank lever
68. Dash-pot plunger
69. Recorder bracket
70. Needle

17. Recorder gate bracket
18. Bolt for 17 and 21
19. Slot in drum flange for 5
20. Gate trigger
21. Recorder drum gate
22. Spring beam roller
23. Roller plate
25. Locating pin
26. Spring beam
27. Brinell ball
28. Tension bar nut
44. Spherical seating
46. Oil plug
47. Reversing nut
48. Worm spindle
49. Operating handle
50. Balance weight hammer
51. Quick-acting handle
52. Cross-head
53. Lock-nut for 54
54. Mercury adjusting screw
55. Mercury adjusting gland nut
71. Screw for 73
72. Cursor retaining spring
73. Index pointer
74. Cursor slide bar
75. Stop pin for 72
76. Glass gauge tube
77. Scale securing screw
78. Rubber connector
79. Mercury steel tube
80. Glass tube clip



restore the zero, drive the automatic controls and recording mechanism, and actuate an automatic load maintainer.

A machine of 4 000 000 lb. capacity installed in the University of California, Berkeley, U.S.A., is illustrated in Fig. 37c.

The Hounsfield Tensometer. The Hounsfield tensometer is a portable universal testing machine which gives a maximum load of 2 tons. It therefore deals with miniature test pieces which, in most materials, give the same results as large test pieces of homologous dimensions with almost uncanny accuracy.

The illustration (Fig. 38) shows a tensile test piece 39 held in chucks or grips 38 which are attached to nose pieces 41 and 42 by means of pins 40. The nose pieces are spherically mounted at 44 to ensure accurate alignment.

Normally the load is applied by hand and it is measured by the deflection of a spring which, in deflecting, causes a mercury piston to displace mercury into a small-bore glass tube adjacent to a graduated scale.

When the handle 49 is turned, the worm gear causes the screw 36—which is prevented from rotating by the key 35 —to move and to apply a load to the test piece.

The other end of the test piece is attached to the tension head 66, which is supported in a bearing at one end and in the spring beam 26 at the other end. A small coupling spring-plate 61 connects the tension bar to the upper end of the lever 62, which is fulcrumed at 65 so that any deflection of the spring beam causes the mercury piston 59 to displace mercury through the steel tube 79 into the glass tube 76 where its movement can be followed on the scale E178, graduated in force, or on the scale E182 graduated in stress.

The screw 58 enables the mercury column to be adjusted to zero before any load is applied.

The movement of the mercury is followed after each increment of load by the cursor index pointer 73; the operator then depresses the needle end of the cursor 70 which punctures the graph sheet on the recording drum.

Extension of the test piece is recorded as follows: The worm wheel 34 drives another gear wheel on the driving shaft 13 which drives the spindle 4 through the train of gears 11, 12 and an idler not shown.

This spindle operates on the drive plate 1 through the pinion 2 which can be set to act on different radii in order to give different rates or magnification ratios for the extension.

Movement of the trigger 20 allows the gate 21 to open and the recorder drum to be removed for changing the graph sheet.

The ends of the spring beam are carried on rollers 22 bearing on hardened plates 23, and the ends of the beams are stepped so that the actual movement of the rollers is negligibly small. When the test piece breaks, the shock is taken by the bell-crank lever 67 which depresses the plunger in an oil dash-pot. The plunger is quite free in its cylinder, and a very light spring keeps the top end of the lever in contact with the tension bar 66.

This spring does not introduce an error as the machine is calibrated with the spring in position.

The reversing nut 47 enables the load to be removed, if required, before the test piece breaks, but normally this nut is removed from the machine; when reversing, the handle 49 disengages the worm and enables the quick acting handle 51 to be used.

If desired, the machine may be motor driven through the spindle 30.

Removal of the nut 28 and the four thumb screws 6/7 enables the spring beams to be changed in a minute, and a series of spring beams of varying thicknesses is supplied to cover a range of loads from 2 tons for testing steel, etc., down to a maximum of 60 lb. for testing silk yarns.

The distance between the nose pieces 41 and 42 can be increased to 22 in., and accessories are provided to enable the following tests to be made: compression, bend, notched-bar, punch shear, cupping and brinell, as well as tensile tests on rods, strips, wires, fabrics and plastics.

Provision is made for determining stress, elongation per cent, and reduction in area per cent of standard test pieces without calculation.

The accuracy of the machine depends on the constancy of a spring, but extensive research on springs of this character has shown that no measurable change takes place for the first million applications of the load, which corresponds to 32 tests a day for 100 years.

The manufacture of the Hounsfield testing machines is handled by Tensometer, Ltd., of Croydon.

Calibration of Testing Machines. To calibrate a testing machine accurately is a matter of some difficulty. The really

satisfactory way is to load the machine with dead weights throughout its entire range. This is seldom possible after a machine has been installed and indirect methods have to be employed.

With single-lever machines the weight of the poise and the length of the short arm of the lever can be verified as follows—

The weight of the poise can be ascertained (*a*) by weighing either with a weighing machine suspended from a crane or by

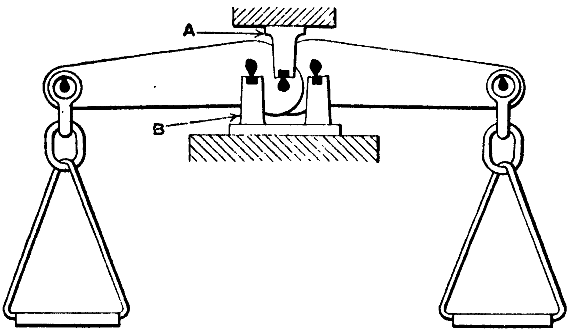


FIG. 39. ARRANGEMENT FOR CALIBRATING MULTIPLE-LEVER TESTING MACHINE
(*Machinery*)

removing the poise and weighing it on a platform machine; (*b*) by the method now to be described.

Balance the beam and adjust the vernier to zero. Hang a known weight w from the beam at a known distance l from the fulcrum. Restore balance by running the poise along the beam. If W is the weight of the poise and L the distance it is moved in order to restore balance, we have by moments

$$W \times L = w \times l$$

hence $$W = wl/L.$$

To check the distance between the knife-edges, balance the beam and adjust the vernier to zero. Hang a heavy weight w in the shackle of the machine and move the poise forwards until equilibrium is restored.

If l_s is the length of the short arm of the lever, by moments

$$l_s = LW/w$$

With multiple lever machines a method frequently adopted is to use proving levers. One form, for use in compression, is

shown in Fig. 39. Two levers are arranged with their fulcra on the part A which bears on the underside of the pulling head. The levers rest on the knife-edge seats in the casting B supported on the platen of the machine. Weights are placed in the scale pans at the ends of the levers and comparison made with the corresponding positions of the poise on the steelyard when the beam is in balance at the respective loads.

The machine is first balanced for the deadweight of the loading device before applying the calibrating weights. The load on the machine is calculated from the leverage ratio, which is usually 10 to 1 or 20 to 1.

Another method of calibration is to employ a standard steel bar and an accurate extensometer. The bar is first tested in a machine of known accuracy and its modulus of elasticity determined. A test is then made on the machine under consideration and the scale readings compared with those calculated by using the modulus of elasticity previously determined.

The American Society for Testing Materials now stipulates that machines up to 50 tons capacity shall be tested at the makers' works by applying dead loads throughout the entire range and the same stipulation applies to calibrating bars. As an alternative to a bar a steel ring may be used. Rings may be obtained fitted with a dial gauge when they are known as *Morehouse rings*. They are stated to be accurate to two-tenths of one per cent between one-fifth and full load capacity. Rings up to 300 000 lb. capacity are obtainable. It is necessary to employ separate rings when calibrating a machine on both rising and falling loads on account of elastic hysteresis in the steel.

A modification of the foregoing method is found in the *Standardizing boxes* made by Messrs. Amsler, Fig. 40. Boxes are made either for tension or compression loading and also in a combined form which permits the box to be used in tension or compression as desired.

The box is a hollow steel cylinder A and is filled with mercury. At one end and on one side of the box the hollow space communicates with a capillary tube C, which terminates in a small bulb. On the other side is a stem provided with micrometer screw plunger E. By displacing with the plunger some of the mercury in the box the end of the mercury column in the capillary tube may be brought to the edge of a datum mark D.

When the box is compressed it shortens and its internal capacity diminishes. This shortening ejects into the capillary tube a volume of mercury equal to the diminution in volume of the interior of the cylinder. The amount of mercury expelled is a measure of the shortening of the cylinder and consequently of the load exerted upon it. The amount so expelled is measured by the micrometer screw.

On commencing the test the micrometer handle is turned until the end of the mercury column in the tube is at the edge

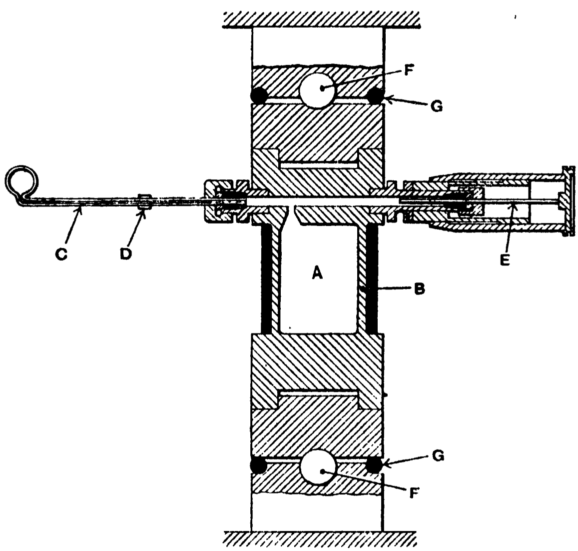


FIG. 40. AMSLER STANDARDIZING BOX
(*Machinery*)

of the datum mark and the scale reading noted. When a load is applied to the box, mercury is forced into the capillary tube. The micrometer is again turned to bring the mercury to the zero mark and the new scale reading observed. The difference between the two readings corresponds to the compressive load on the box.

The mercury meniscus is always restored to the same point before each reading of the micrometer. The results are thus independent of variations in the diameter of the glass tube.

The box is insulated at *B* to prevent rapid changes of

temperature from causing an alteration of the mercury column. To prevent the standardizing box from being stressed eccentrically it is necessary to support the box so that it can adjust itself freely to the axis of the machine. It is also necessary to take precautions to ensure that the box does not upset in case one of the compression plates is not guided but can yield laterally. If neither compression plate can yield laterally it is advisable to apply the load through balls *F* placed between the seatings. The rubber ring *G* prevents the box from falling over after it is completely unloaded.

If one of the compression plates can yield laterally, as in the Amsler machine, it is necessary to dispense with one of the steel balls and the accompanying seating and to place the standardizing box on the fixed compression plate of the machine while retaining the other ball and its seating.

The loads recorded by the weighing apparatus of the machine being calibrated are compared with a scale of loads engraved on the box.

Compression boxes are made for machines up to 500 tons and tensile boxes for machines up to 300 tons capacity.

The boxes can be used equally well with vertical or horizontal machines and each box is guaranteed to an accuracy of within $\frac{1}{2}$ per cent for all loads above one-tenth the capacity of the box.

Calibration of a 600 000 lb. Riehle Machine. Another method of calibrating a machine throughout its entire range consists in the use of dead weights together with a calibrating bar. A 600 000 lb. Riehle machine was calibrated by this method at the University of Illinois by using two 20 000 lb. weights and a steel bar, 19 ft. long and 4 in. diameter, carrying a strainmeter of 100 in. gauge length.

One complete division of the micrometer dial of the strainmeter represented 0·00005 in. stretch.

The procedure was as follows. After balancing the machine at zero load the weights were lowered on to the table by means of jacks. The machine was again balanced and the error at 20 000 lb. noted.

With the poise weight at the position found, the weights were lifted off the table until the beam was once more in balance. The reading of the strain gauge was noted and the weights again lowered on the table. This dead load relieved the pull in the bar to a small extent and caused the strainmeter reading to fall

slightly. Further load was applied through the straining gear to bring the meter back to its former reading. The load on the machine now consisted of 20 000 lb. dead load and 20 000 lb. pull in the bar, making a total of 40 000 lb.

After balancing the beam and determining the error at this load the procedure was continued in increments of 20 000 lb. until full load was reached.

With this method it is not necessary to evaluate the modulus of elasticity of the bar. All that is required is that a loading should be reproduced accurately. Any error in a reading is carried forward throughout the series, but some counterbalancing of such errors should occur as the calibration proceeds.

The maximum error in the machine tested, 0·15 per cent, occurred at 50 000 lb. load. At 300 000 lb. the error was negligible and at full load was less than 0·1 per cent.

Jakeman's Method of Calibration. The method of calibrating a single-lever testing machine described on page 66 has been elaborated by Jakeman. Two pieces of angle iron each with a V-groove milled along its whole length and having a hole drilled at its centre O, Fig. 41 (*a*) and (*b*), are bolted to the ends of the steelyard. The angle irons are of such a length that the lines *AA* and *BB* are clear of projections on the machine. The V grooves provide seatings for supporting the scale pans P_1 and P_2, Fig. 42. The stem of each scale pan is attached to a short length of steel rod which rests in the groove in the angle iron, Fig. 41 (*b*).

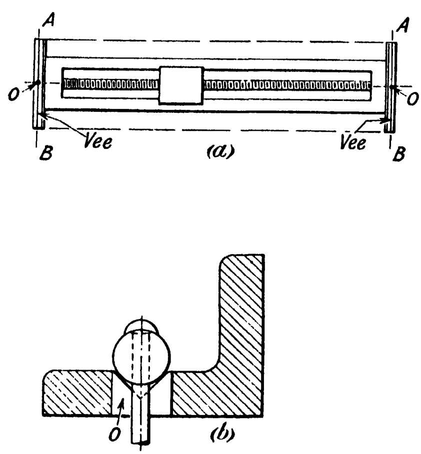


FIG. 41. JAKEMAN'S METHOD OF CALIBRATION

(*a*) Attachment of angle irons to weigh-beam.
(*b*) Method of suspending scale pans.

The lengths *AA* and *BB* are measured by means of an accurate steel tape, the mean of the two measurements giving the distance *L* between the suspensions P_1 and P_3.

For the calibration two or three 100 lb. weights, a number of

2 lb. weights, and a set of weights from 1 lb. to 0·001 lb. are needed.

An additional scale pan P_3 is suspended from the loading shackle of the machine.

To determine the weight of the poise (w).

As many of the 100 lb. weights as convenient are placed on the pan P_1 and the lever balanced by moving the poise. The

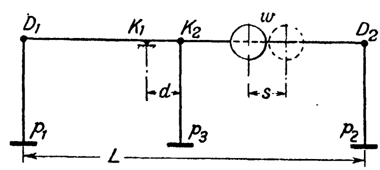


FIG. 42. JAKEMAN'S METHOD OF CALIBRATION

100 lb. weights are now moved from P_1 to P_2 and the distance S through which the poise has to be moved to secure balance is measured.

If W be the total weight moved from P_1 to P_2 then

$$w = (\mathrm{W} \times \mathrm{L})/S$$

To determine the knife-edge distance (d).

The small weights are all placed on P_2 and the beam balanced. The poise is then wedged in position so that it cannot move. The large weights are placed on P_3 and the beam balanced by moving weights from P_2 to P_1. If m is the sum of the weights removed from P_2 to P_1 and W′ the total weight placed on P_3,

$$d = (\mathrm{L} \times m)/\mathrm{W}'$$

The length l on the scale representing 1 ton can be checked without measuring the distance L, for

$$l = \frac{d \times 2\,240}{w} = \frac{\mathrm{L}m}{\mathrm{W}'} \times \frac{2\,240S}{\mathrm{WL}}$$

The method may be used to check the knife-edge distance under load using a large weight on P_3. In order to obtain sufficient sensitivity, the motion of the lever must be appreciable when moving the small weights from P_2 to P_1. Jakeman suggests that a load of the order of 50 tons should be applied through a railway coupling. The ram must be blocked in position if the machine is hydraulically loaded.

As the sensitivity of the machine is less under these conditions several observations should be taken.

Dynamic Calibration of Testing Machines. It by no means follows that the indications of a testing machine, given under static calibration, will be repeated under operating conditions where a test piece is loaded continuously to failure at a specified rate of loading.

A recent investigation by Wilson and Johnson* has shown that considerable errors may occur under such conditions. A number of machines were tested, including a beam-and-poise screw-power machine and several hydraulically loaded machines indicating either by a pendulum dynamometer or by a pressure gauge.

The calibrating apparatus, of 25 000 lb. capacity, consisted of a steel ring containing an electrical contact device. Its sensitiveness was less than 2 lb. for any load within the maximum.

The indicated loads of the machines for dynamic loadings were obtained by photography with a motion picture camera, this indication and a neon lamp in the circuit of the calibrating device being photographed together. The contact was arranged so that the lamp ceased to glow at a predetermined load. Any error so found was added algebraically to the error obtained under static calibration.

In the beam-and-poise machine, in which the poise was hand-operated, it was found that the errors due to overrunning the poise masked any error due to the rate of loading. The error from this cause was proportional to the rate of loading, and the indicated loads were greater than the applied loads.

In the hydraulic machines considerable errors were in some cases found to be due to the use of the maximum pointer on the gauge, while in other cases the error was too small to be measured by the method adopted.

With one or two exceptions, the error was independent of the load and in the hydraulic machines the indicated load was less than the applied load. The additional errors in several instances exceeded the tolerances specified by the American Society for Testing Materials.

Unfortunately, the results of such tests cannot be used to correct the load readings of other machines of the same type.

* National Bureau of Standards, U.S.A. Research Paper R.P. 1009.

CHAPTER IV

TENSION AND BENDING TESTS

The Load-extension Diagram for Mild Steel. If a bar of wrought iron or mild steel be tested in tension a graph showing the relation between the load and extension, or between stress

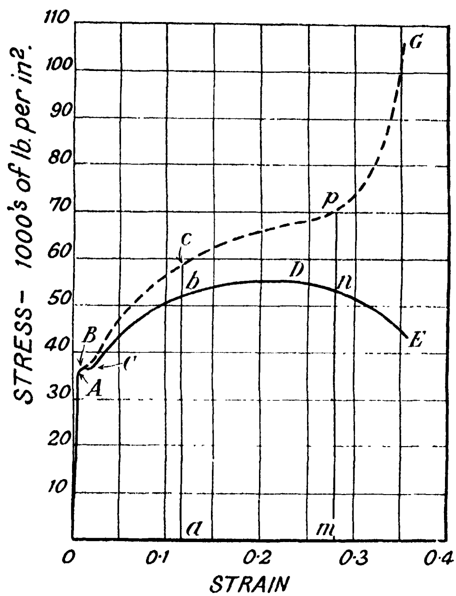


FIG. 43. STRESS-STRAIN CURVE FOR MILD STEEL IN TENSION

and strain, will be, generally, of the form shown by the full line curve in Fig. 43.

The extension will be found to be proportional to the load over a considerable range OA, after which, as the load is gradually increased, the graph will deviate from a straight line and for very little greater load the extension will proceed rapidly without any further rise of load occurring. This is indicated by the portion BC of the curve. Actually, a sudden

drop of the load occurs at this stage owing to the rapidity with which the specimen extends. The reduction of load is made visible by a drop of the weighbeam of the testing machine, or, if an autographic record be taken, the graph will usually show a drop at this point to an extent depending on the sensitiveness of the recording gear. A record taken on a commercial machine may show a reduction of 5 per cent of the load, but with special apparatus under laboratory conditions a drop of 27 per cent has been recorded.

The point B at which this sudden yield occurs is termed the *yield point* of the material.

In wrought iron and mild steel the limit of proportionality and the elastic limit practically coincide, but this is not generally so with other materials or with a material that has been overstrained.

With further application of the straining action the test piece will withstand still higher loads, but stress and strain are no longer found to be proportional. The graph follows some such line as CD until the maximum load is reached. During this plastic stage the cross-section of the material diminishes in about the same proportion as the length increases, and on passing the maximum load a sudden local stretching takes place over a short length of the test piece and a waist or neck is formed.

This greatly reduced area is insufficient to sustain the load and it will be found that, to preserve balance of the steel yard, the poise must be run back towards zero.

From D the curve falls until fracture of the test piece occurs, indicated by the point E.

The actual stress at fracture is, however, much higher than that corresponding to the maximum load since it is given by

$$\frac{\text{Load}}{\text{Actual area of cross-section}}$$

and the proportionate reduction in section is considerably greater than the reduction in load. The graph, therefore, exhibits only a nominal indication of the stress at any point during the plastic stage.

It is difficult to obtain the complete load-extension curve by direct observation but, supposing the curve obtained and plotted as a stress-strain diagram, the corrected curve, shown dotted in the figure, may be plotted as follows.

Assuming that the volume of the specimen keeps constant, the ordinate

$$ac = ab \times \frac{\text{stretched length}}{\text{original length}},$$

uniform contraction accompanying the extension.

The relation given enables the true stress-strain curve to be plotted up to the point at which a waist begins to form, but from this point onwards the relation ceases to be true.

However, for any point n beyond the point of maximum load the true stress is given by

$$mp = mn \times \frac{\text{original area}}{\text{reduced area}}.$$

The final point G may be obtained with some accuracy, as it is a fairly easy matter to measure the reduced diameter and the total elongation after fracture.

In commercial practice the breaking stress is taken to be the nominal value

$$\frac{\text{Maximum load}}{\text{Original area of cross-section}},$$

and is termed the maximum stress or *ultimate stress*.

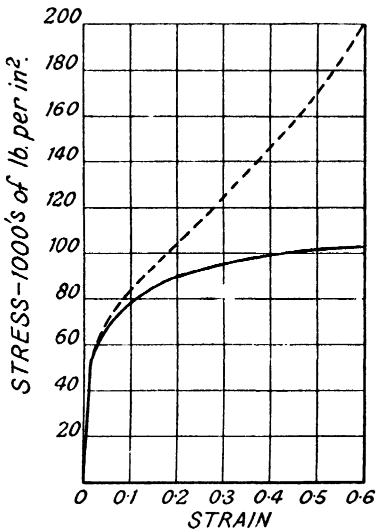


FIG. 44. STRESS CURVE FOR WROUGHT IRON IN COMPRESSION

A curve for wrought iron in compression is shown in Fig. 44. There is no definite ultimate crushing stress, nor is the yield point so well defined as in tension. The curve of actual stress becomes more nearly parallel to the stress axis as the load increases.

Elongation, Contraction of Cross-section. A bar of ductile material, before it fractures, will draw out into a waist and fracture will occur at the reduced section.

If a bar, previously marked off into 1 in. lengths, be tested in tension and if, after fracture, the extension of each inch length be measured, it will be found that the unit extension is greatest in the neighbourhood of the fracture and least at points farther away. Moreover, if the extensions be plotted as ordinates

at the centres of the respective inch divisions, a curve drawn through the plotted points will be fairly symmetrical about the position of fracture (Fig. 45). It will be noted that the extension is nearly constant except in the immediate vicinity of the fracture.

The percentage extension, a criterion of ductility, depends upon the gauge length adopted. Fig. 46 is obtained by plotting

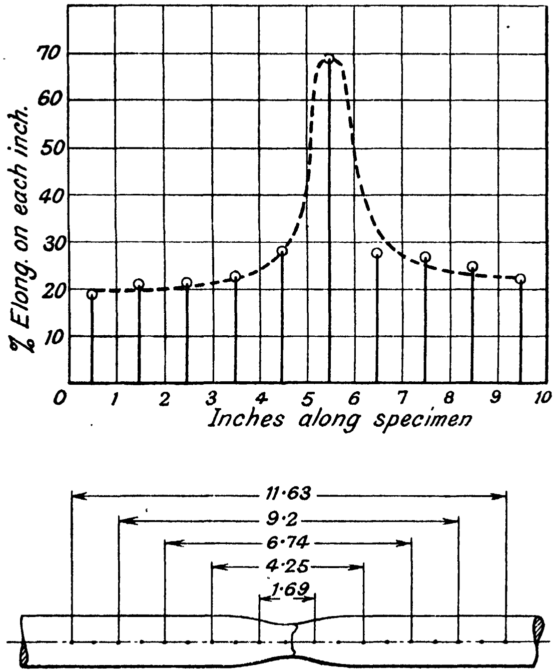


FIG. 45. VARIATION OF UNIT-EXTENSION ALONG GAUGE LENGTH

the percentage extensions observed in a test on a mild steel bar on gauge lengths of 1, 3, 5, 7, and 9 in., all of which include the fractured section. The extension on a gauge length of 2 in. is 51 per cent, while on a gauge length of 8 in. it is only 26 per cent. This shows clearly the necessity for stating the gauge length when specifying the percentage elongation. Both the percentage elongation and the percentage reduction in area of cross-section are regarded as criteria of ductility.

The total elongation of the fractured test piece is made up

of the local extension in the region of fracture together with the more uniform extension over the remainder of the gauge length.

The local extension occurs mainly after the maximum load is reached and at the point where the neck or waist forms. The length over which the local extension exists depends to some extent upon the area of the cross-section and is approximately proportional to the square root of that area. Unwin has shown that the relation between the percentage elongation e and the dimensions of the test piece may be represented with fair accuracy by an equation of the form

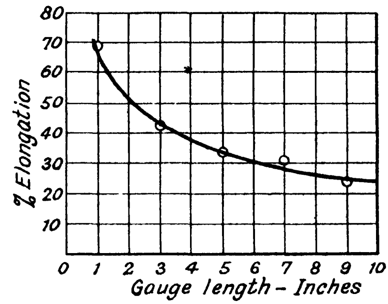


FIG. 46. PERCENTAGE ELONGATION VARIES WITH THE GAUGE LENGTH

$$e = (c\sqrt{a})/l + b$$

where $(c\sqrt{a})/l$ represents the local extension and b the general extension; a being the area of cross-section and l the gauge length.

Average values for the constants c and b are—

Material	Values of the Constants	
	c	b
M.S. plates, not very thick	70	18
Gun metal (cast)	8·3	10·6
Rolled brass	101·6	9·7
Rolled copper	84	0·8
Annealed copper	125	35

Proportions of Test Pieces. The percentage elongation depends also on the form of the test piece and it was suggested by Barba in 1880 that for strictly comparable results, test pieces should be similar.

In the case of cylindrical test pieces this leads to the relation l/d = a constant. In Germany this ratio is usually taken as 10, which makes $l = 11{\cdot}3\sqrt{a}$, where a is the cross-sectional area in cm.^{2} and l is the length in cm.

The values adopted by the British Standards Institution are—

d (in.)	a (in.^{2})	l (in.)
0·564	0·25	2
0·798	0·5	3
0·977	0·75	3·5

The above values correspond approximately to $l = 4\sqrt{a}$.

In America the relation $l = 4{\cdot}5\sqrt{a}$ is adopted.

In the testing of plates, where test pieces are rectangular in section, adherence to the similarity law is attended with difficulties and hence considerable departure from the theoretic law is made.

For instance, the B.S.I. test pieces for plates are of the form shown in Fig. 47, and to lessen the cost of production three

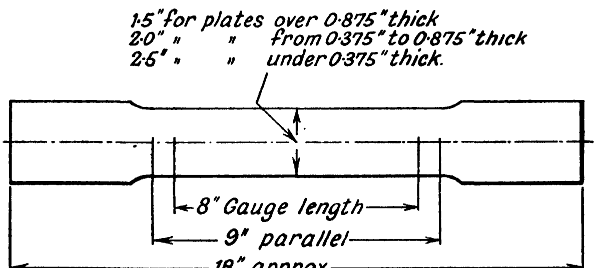


FIG. 47* FORM OF BRITISH STANDARD TEST PIECES FOR PLATES

widths only are specified. The width of 2 in. was selected for the sake of uniformity and as the percentage elongation was found to be less for thick than for thin plates the other widths were chosen in order to slightly favour the thicker plate.

To secure the same percentage elongation with the same material, for round specimens the gauge length is made eight times the diameter and the parallel portion nine times the diameter.

The specification for cylindrical test pieces for forgings,

* Abstracted by permission from British Standard Specification 18—*Forms of Tensile Test Pieces*, copies of which can be obtained gratis from the British Standards Institution, 28 Victoria Street, London, S.W.1.

axles, etc., permits various lengths and diameters to be employed. Particulars of these are given in Table III, which has reference to Fig. 48.

The form of the ends of the test piece is a matter of some choice, the largest diameter and particulars of the thread on the screwed part not being laid down by the specification.

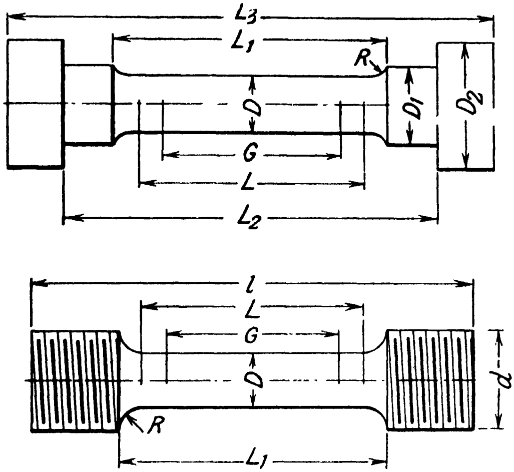


FIG. 48.* FORMS OF CYLINDRICAL TEST PIECES

It not infrequently happens that the material available makes it necessary to work to close dimensions with regard to the ends of the test pieces. Particulars of the Hounsfield system of fundamental dimensions and tolerances are given in Fig. 49.

Another form of test piece is shown in Fig. 50. The extra length of plain portion permits a Brinell test to be made on specimen.

The position of the fracture on the test bar will influence to some extent the percentage elongation. If fracture occurs within the middle third of the gauge length the resulting value of the percentage elongation may be accepted, but if outside this limit a correction should be made if it is desired to obtain strict comparisons.

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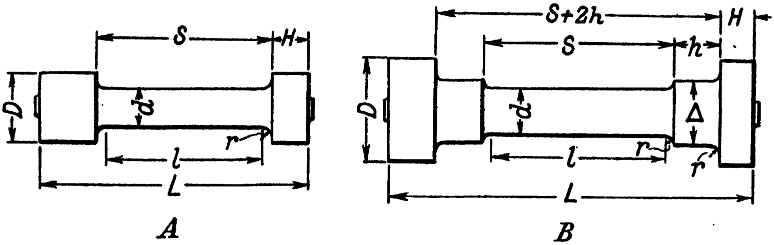

FIG. 49. STANDARD TEST SPECIMENS

		Dimensions	Tolerances
SPECIMEN A	l	$4\sqrt{(\text{area})} = 3{\cdot}5449d$	(On d): $+ 0{\cdot}0025d$, $- 0{\cdot}0012d$
	r	$0{\cdot}2d$	$+ 0{\cdot}05d$, $- 0{\cdot}025d$
	S	$l + 2r = 3{\cdot}95d$	Use only "go" gauge, about 0·001 in. short
	H	$0{\cdot}8d$ (min.)	$+ 0{\cdot}4d$, $- 0$
	L	$S + 2H = 5{\cdot}55d$ (min.)	
	D	$1{\cdot}5d$ (min.)	$+ 0{\cdot}1d$, $- 0$
SPECIMEN B	l	$4\sqrt{(\text{area})} = 3{\cdot}5449d$	(On d): $+ 0{\cdot}0025d$, $- 0{\cdot}0012d$
	r	$0{\cdot}2d$	$+ 0{\cdot}05d$, $- 0{\cdot}025d$
	S	$l + 2r = 3{\cdot}95d$	Use only "go" gauge, about 0·001 in. short
	h	d (approx.)	
	H	$0{\cdot}7d$ (min.)	$+ 0{\cdot}3d$, $- 0$
	L	$S + 2(H + h) = 7{\cdot}35d$ (min.)	
	Δ	$(\sqrt{2})d$	$+ 0$, $- (0{\cdot}05d + 0{\cdot}004$ in.)
	D	$(\sqrt{2})\Delta = 2d$	$+ 0{\cdot}25d$, $- 0$

Note. A large tolerance is given to r to enable standard round rod to be used in the lathe tool. The length l is gauged by using the length S, but in obtaining S the actual value of r is taken into account, in order to make l correct.

In the Figures the left-hand ends are drawn to upper limits and the right-hand ends to lower limits.

Small "pips" are left on the ends of the test pieces to facilitate measurement of the total extension by means of a micrometer. If centre dots are placed at the gauge points to receive an extensometer, the stem length S should be increased by the difference between G and L in Fig. 48.

Timoshenko gives the following method of making the correction.

If the gauge length has been divided into n parts of equal length before testing and if l_t = the total extension required and x = the extension on the division containing the fracture;

then, if fracture occurs outside the middle third and the number of divisions n is even, the measurements being made on the longer part of the test piece, the extension

$$l_t = x + \text{extension on } (n/2 - 1) \text{ divisions} + \text{extension on } n/2 \text{ divisions.}$$

If n is odd

$$l_t = x + \text{extension on } \tfrac{1}{2}(n - 1) \text{ divisions} + \text{extension on } \tfrac{1}{2}(n - 1) \text{ divisions}$$

where the measurement is again made on the longer part of the test piece.

According to a German rule a test piece of 200 mm. gauge length is divided into 20 equal parts. If fracture occurs outside the middle third the division marks are numbered on

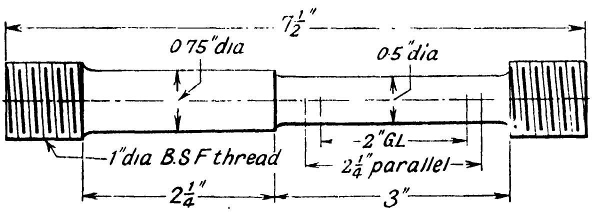

FIG. 50. ALTERNATIVE FORM OF TEST PIECE

either side of and away from the fracture, 0 to n on the shorter portion, and 1 to 10 on the longer portion. Then the required extension

l_t = length of fractured division + length between 0 and n + length between 1 and 10 + length between n and 10 on the long portion.

For a gauge length of 100 mm. the gauge length is divided into ten equal parts; five being then substituted for ten in the foregoing rule.

Certain types of extensometer are pinched on to the specimen by means of pointed set screws. The small centre pops formed in the test piece at the gauge points tend to cause fracture to occur there rather than in the middle of the gauge length. Should fracture occur at a gauge point the test should be disregarded. A test is generally considered to be satisfactory if fracture occurs within the gauge length and not nearer to a gauge point than a distance equal to $\sqrt{(\text{area of cross-section})}$.

TABLE III

Dimensions of Cylindrical Test Pieces

Diameter D	Area of Cross-section	Gauge Length G	Length of Parallel L	Length between Shoulders		Overall Length		D_1	D_2	d	Radius R
				L_1	L_2	L_3	l				
In. 0·564	In.^{2} 0·25	In. 2·0	In. 2·25	In. 3·0	In. 4·25	In. 5·25	In. 4·5	In. 0·75	In. 1·25	$\frac{3}{4}$ B.S.F.	As large as convenient
0·798	0·50	3·0	3·375	4·0	5·25	6·5	6·5	1·0	1·5	$1\frac{1}{4}$ B.S.W.	
0·977	0·75	3·5	4·0	5·0	6·25	8·0	8·0	1·25	1·75	$1\frac{1}{2}$ B.S.W.	

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To prevent fracture at a gauge point, brittle materials are tapered two- or three-thousandths of an inch from the gauge point towards the centre, the middle portion of the gauge length being left parallel.

Forms of test pieces for brittle materials are shown in Fig. 51.

Proof Stress. The employment of the terms "limit of proportionality," "elastic limit," and "yield point," has in the past occasioned some confusion through the precise significance of the terms not being realized or borne in mind. Apart from any question as to its utility as a characteristic on which to base the suitability of a material, the limit of proportionality is difficult to determine with accuracy and in some materials is not in evidence.

The elastic limit is impracticable of determination in ordinary commercial testing and the yield point, while well defined in the mild steels, is ill defined in others and absent in some and in the non-ferrous alloys.

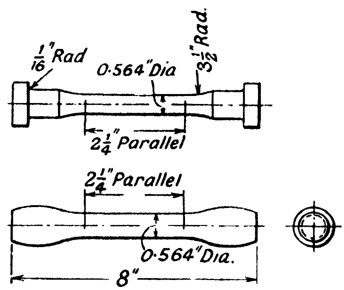

FIG. 51.* FORMS OF TEST PIECES FOR BRITTLE MATERIALS

Owing to these difficulties the term "proof stress" came to be used in specifications especially when the material in question was such as did not show a well-defined yield. On a given intensity of stress being applied to a test piece for a given length of time the resulting permanent set of the material was specified not to exceed a pre-assigned amount—1 per cent or 0·5 per cent of the elongation under a given load.

Proof stress now figures prominently in the specifications of many authorities, and is defined as the stress which is just sufficient to produce a permanent elongation equal to a specified proportion of the gauge length.

When a precise value of the proof stress is required a stress-elongation or load-elongation curve must be plotted. Often, however, the precise value is of little moment, it being of more importance to know that the proof stress is not below a given value, or that it lies between specified limits.

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The method of determining proof stress by means of the load-elongation curve is illustrated in Fig. 52, which represents the results of a test on a steel specimen of 2-in. gauge length.

ABC is the load-extension curve obtained by plotting corresponding values of load and extension. The abscissa

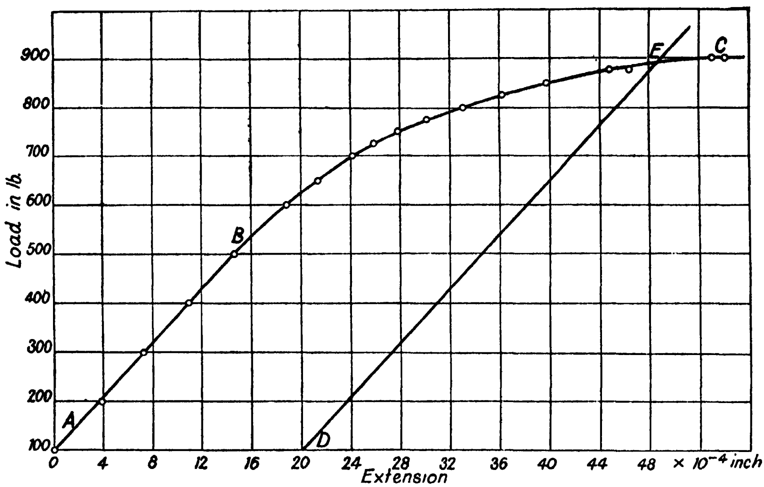

FIG. 52. METHOD OF DETERMINING PROOF STRESS FROM THE LOAD-EXTENSION CURVE

gives the extensometer readings in scale divisions which on multiplication by 10^{-4} yield extensions in inches.

The extension specified at the proof stesss is 0·1 per cent of the gauge length, that is, $0 \cdot 1 \times 2/100 = 0 \cdot 002$ in., corresponding to twenty divisions on the extensometer dial. Readings are commenced from a small initial tensioning load, in this instance 100 lb.

Having plotted the graph, a straight line *DE* is drawn parallel to the part of the curve representing the proportional extension at a horizontal distance from it equal to the specified extension of 0·002 in., namely, at the point marked 20 on the abscissa. The point *E* where this line cuts the load-extension curve give the proof load, in this case 875 lb. The corresponding proof stress is 19·32 tons per square inch.

Various methods in vogue for determining whether or not a material satisfies proof stress requirements avoid the necessity of plotting a stress-strain curve. They are to be found in the specifications of several authorities.

One, known as the four-point method, enables a close determination of the value of the proof stress to be made when the upper and lower limits for the proof stress are specified. The limits are based on previous knowledge of the behaviour of the material.

The procedure is as follows—

1. Referring to Fig. 53, a tensioning stress OA of about 20 per cent of the specified minimum proof stress is applied to

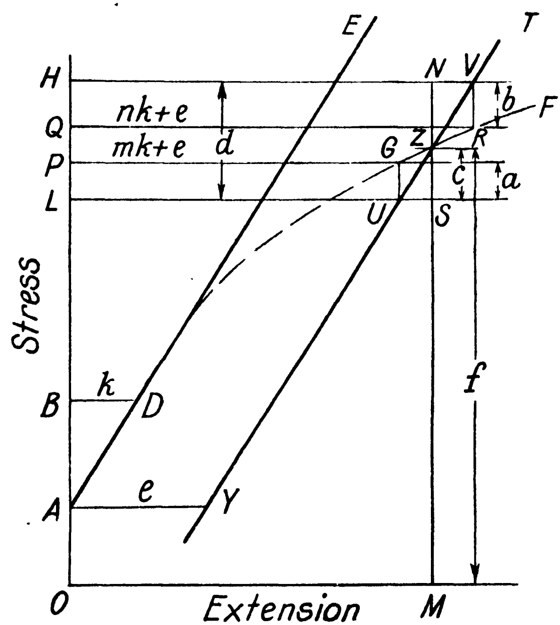

FIG. 53. FOUR-POINT METHOD OF DETERMINING PROOF STRESS

the test piece and the extensometer set to zero. The ordinates OL and OH represent the lower and higher values of the proof stress.

2. The stress is increased to an amount represented by OB where AB is so chosen that AL/AB and AH/AB are whole numbers, m and n, say. This is not essential but is merely for convenience of calculation.

3. The extensometer reading $BD = k$ is noted.

4. The load is increased until the extensometer reading $PG = mk + e$, where e is the specified percentage extension:

0·002 in. for 1 per cent on 2 in. gauge length. (On some extensometers, such as the Gerard, this corresponds to twenty divisions.) The load corresponding to this extension is observed.

5. The load is further increased until the extensometer reading is $QR = nk + e$, and the load again noted.

If the foregoing observations are plotted the points G and R will be obtained. These lie on the stress-extension curve ADF. By drawing YT parallel to AD and distant from it an amount $AY = e$, and joining GR as indicated in the figure, the proof stress is obtained as $MZ = f$. The stress f is made up of the lower limit of proof stress OL plus the amount c; that is, $f = OL + c$.

From the pairs of similar triangles ZNV, ZSU and ZRV, ZGU,

$$RV/GU = ZV/ZU = ZN/ZS.$$

Let $GU = a$, $SZ = c$, $RV = b$, $SN = d$, $ZN = d - c$, as in the figure. Then it follows that—

$$\frac{d-c}{c} = \frac{b}{a}$$

Adding 1 to both sides gives

$$\frac{d-c}{c} + 1 = \frac{b}{a} + 1$$

or

$$\frac{d-c+c}{c} = \frac{b+a}{a}$$

whence

$$c = \frac{ad}{a+b}$$

Thus the amount to be added to OL, the lower limit of proof stress, can be determined without plotting a diagram.

Determination of the Modulus of Elasticity. The modulus of elasticity is sometimes obtained in the following manner. A load is applied to the test piece sufficient to produce a small permanent set and the reading of the extensometer observed. The load is then nearly all removed and the extensometer reading again observed. The value of the applied stress divided by the difference between the extensometer readings, expressed as a strain, is taken as the modulus of elasticity. The merit of the test lies in the rapidity with which it can be carried out.

The modulus of elasticity, however, is preferably obtained from a graph plotted from the recorded loads and extensions as

in Fig. 54, which is from a test on a specimen 0·56 in. diameter and 2 in. between gauge points from a steel casting. The limit of proportionality, 10 300 lb., corresponds to a stress of 19·56 tons per in.^{2}

$$\text{The modulus of elasticity E} = \frac{\text{load} \times \text{length}}{\text{area} \times \text{extension}}.$$

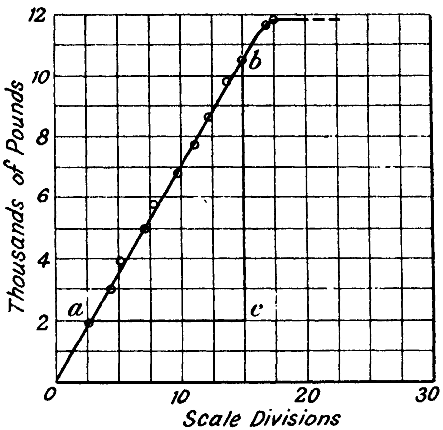

FIG. 54. USE OF GRAPH IN DETERMINING YOUNG'S MODULUS

The value of one scale division on the extensometer used was 1/5 000 in. From the graph, taking two points *a* and *b* on the straight portion, we find that the extension *ac* scales 12·1 instrument divisions and the load *cb* scales 8 500 lb.

The extension is

$$12{\cdot}1 \times \frac{1}{5\ 000} = 0{\cdot}00242 \text{ in.}$$

Hence

$$\text{E} = \frac{8\ 500 \times 2}{(\pi/4) \times (0{\cdot}56)^2 \times 0{\cdot}00242}$$
$$= 28\ 500\ 000 \text{ lb. per in.}^2$$

In this example the values of the load and extension might have been read off from one point on the graph as this passes through the origin. Such is not always the case, as corresponding to zero reading of the instrument a slight load on the

specimen may have to be imposed in order to ensure that the grips are biting if specimens are being tested in wedge grips. As a fair curve drawn the plotted points may not, in general, pass through the origin, the slope of the graph should always be determined by taking two selected points on the line.

Form of Report for Tensile Tests. A form of report for tensile tests is given below.

REPORT OF TENSILE TEST FOR ....

Date of Test ....

Particulars of Specimen . . . *Flat Machined Steel* . . . . .

No. of Specimen ....

Mark of Owner ....

Original dimensions—		
Length between gauge points	8	in.
Breadth	1·546	in.
Thickness	0·238	in.
Diameter	—	
Original area of cross-section . $1{\cdot}546 \times 0{\cdot}238 =$	0·368	in.^{2}
Limit of proportionality—		
Total load	—	tons
Stress	—	tons per in.^{2}
Yield—		
Total load	9·3	tons
Stress	25·27	tons per in.^{2}
Maximum load	14·46	tons
Ultimate stress	39·74	tons per in.^{2}
Load at rupture—		
Total	—	tons
Stress	—	tons per in.^{2}
Dimensions of cross-section at fracture . $1{\cdot}213 \times 0{\cdot}15$		in.
Area of cross-section at fracture . $1{\cdot}213 \times 0{\cdot}15 =$	0·182	in.^{2}
Contraction of area . $0{\cdot}368 - 0{\cdot}182 =$	0·186	in.^{2}
Percentage of original area . $0{\cdot}186 \times 100/0{\cdot}368 =$	50	
Young's modulus	13 500	tons per in.^{2}
Elongation in 8 in.	1·25	in.
Percentage on original length . $1{\cdot}25 \times 100/8 =$	15·6	

Remarks—

Extensometer reading at 2 tons per in.^{2} . . . . 19·3

$= 0{\cdot}00386$ in.

Extensometer reading at 17 tons per in.^{2} . . . 65·1
= 0·01302 in.

Extensometer reading at 2 tons per in.^{2} . . . 20·9
= 0·00418 in.

Permanent set on removal of load 0·0418 − 0·00386
= 0·00032 in.

Young's modulus = (17 − 2) × 8/(0·01302 − 0·00418)
= 13 575 tons per in.^{2}

Transverse Tests on Cast Iron. The most convenient test to apply to cast iron is the *transverse test* in which a bar of round or rectangular section is supported at its ends and loaded in the centre until fracture occurs. The load needed to cause fracture diminishes with increase of span, and hence the apparatus required for making the test may be comparatively light in structure and inexpensive. This makes the test particularly convenient for foundry and workshop use. On the other hand the test can be made equally well, or better, in the ordinary tension and compression testing machine. In addition to measuring the load it is customary to measure the deflection as this affords some indication of the toughness of the material. To measure the deflection with some accuracy an Ames dial or similar instrument should be used.

Tensile tests of cast iron are frequently called for, and both the transverse and the tensile test have been standardized by the British Standards Institution.

Formerly, test bars for the transverse test were 2 in. × 1 in. × 42 in. long and were tested on edge by applying a load at the centre of a span of 36 in. The B.S.I. specification, however, has been brought more into line with the German and American Standards and test bars are now made to the following dimensions—

TABLE IV

STANDARD TEST BARS FOR TRANSVERSE TESTS

Test Bar	Diameter (in.)	Overall Length (in.)	Main Cross-sectional Thickness of Casting Represented (in.)
S	0·875	15	Not exceeding $\frac{7}{8}$
M	1·2	21	Over $\frac{7}{8}$ and not exceeding 2
L	2·2	21	Over 2

Bars which exceed the standard diameters by more than 0·1 in. must be turned down to the standard dimensions. Certain minimum test requirements have to be met if bars vary more than 0·1 in. from the standard dimensions. (See B.S.S. No. 321.) The form of test piece for tensile tests is shown in Fig. 55, to which Table V refers. The corresponding loadings are given in Table VI.

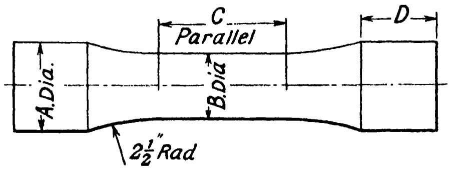

FIG. 55.* TENSILE TEST PIECE FOR CAST IRON

Modulus of Rupture. The transverse strength of cast iron is often gauged by the *modulus of rupture*. This quantity is the ratio of the maximum bending moment at failure to the modulus of the section (page 9). The modulus of rupture is thus the greatest stress that would have obtained had the material obeyed Hooke's law up to the point of fracture. If this condition existed the value of the maximum stress would be independent of the shape and size of the section. However, since the linear relationship between stress and strain ceases

TABLE V
STANDARD TEST PIECES FOR TENSILE TESTS

Test Bar	Dimension (in.)				Main Cross-sectional Thickness of Casting Represented (in.)
	A	*B*	*C*	*D*	
S	0·875	0·564	2	1	Not exceeding ¾
M	1·2	0·798	2	1	Over ¾ and not exceeding 2
L	2·2	1·785	2	1	Over 2

Test bars are to be cast as parallel bars of the diameter given in *A* and machined to the dimensions *B* and *C*.

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TABLE VI

VALUES OF THE MODULUS OF RUPTURE FOR BARS UNDER STANDARD CONDITIONS

Test Bar	Distance between Supports (in.)	Transverse						Tensile	
		Minimum Load in lb.		Minimum Deflection (in.)		Modulus of Rupture Tons per in.^{2}		Minimum Tons per in.^{2}	
		Grade		Grade		Grade		Grade	
		I	II	I	II	I	II	I	II
S	12	1 185	960	0·12	0·1	24·1	19·6	12	10
M	18	1 950	1 600	0·15	0·12	23·1	18·9	11	9
L	18	10 000	8 950	0·12	0·1	19·2	17·2	10	9

long before the point of fracture is reached, the modulus of rupture must be regarded "as a convenient way of expressing the results of transverse tests without giving full details of the bar dimensions."

If L is the distance between the supports in inches;
W the load applied at the centre of the span in tons;
Z the modulus of the section;
f the maximum stress induced, tons per in.^{2}

then
$$f = WL/4Z$$

and if W is the load when fracture occurs the corresponding value of f is the modulus of rupture.

For a rectangular bar $z = \dfrac{\text{breadth} \times \text{depth}^2}{6}$

For a round bar $z = 0{\cdot}0982\ (\text{diameter})^3$

Values of the modulus of rupture for bars under standard conditions are given in Table VI. With the older standard it was usual to specify that on a 36 in. span the bar should sustain a load of 30 cwt. at the centre with a deflection of not less than 0·625 in. The modulus of rupture is affected by the method and conditions of casting, by the shape of the section, and by the span of the test piece. Rough bars are stronger than planed bars, and with bars of similar proportions the modulus is lower as the section is larger. A wide bar tends to give a higher and a deep bar a lower modulus of rupture. Fig. 56, from Bach's *Elastizität und Festigkeit*, indicates how greatly the modulus of rupture is influenced by the shape of the section.

The ratio of the modulus of rupture to the tensile strength varies from 1·25 to 3; the higher value occurring with irons of lower tensile strength.

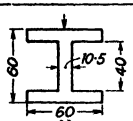
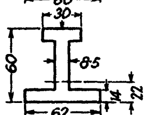
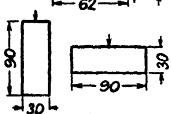
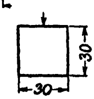
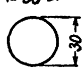
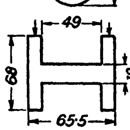
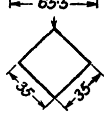
Shape of Section Dimensions in millimetres	Ratio of Modulus of Rupture to Tensile Strength
	1·45
	1·52
	1·75
	1·73
	2·12
	2·14
	2·35

FIG. 56. RELATION OF MODULUS OF RUPTURE TO SECTION (BACH)

When a test bar is not of the standard dimensions the load that would produce the same stress in a bar of the same material of standard dimensions is termed the *equivalent load.* The equivalent load is calculated by the usual theory.

For a bar of rectangular section, if W be the actual load on a bar of breadth b and depth d, the load W′ that would produce the same stress in a standard bar of breadth b' and depth d' is given by

$$W' = Wb'd'^2/bd^2$$

The load W_1 that would produce the same deflection as in the given beam is from the relation

$$\delta = Wl^3/48EI = Wl^3/4Ebd^3$$

obtained as

$$W_1 = W(b'd'^3/bd^3).$$

But

$$W_1 = W'\frac{bd^2}{b'd'^2}\cdot\frac{b'd'^3}{bd^3} = W'\frac{d'}{d}$$

and hence is not equal to the load required to produce the same stress.

The deflection produced by the "equivalent" load in the standard bar is

$$\delta' = \frac{W'bd^3}{b'd'^3W}\,\delta = \frac{b'd'^2}{bd^2}\cdot\frac{bd^3}{b'd'^3}\,\delta = \frac{d}{d'}\,\delta$$

This relation holds equally for bars of round section.

Crushing and Shear Tests. Crushing tests on cast iron are made on cylinders or prisms in which the ratio of the height

to the least lateral dimension is from 1 to 3. A suitable form of test piece is shown in Fig. 57.

The crushing strength as given by tests varies from 30 to 70 tons per square inch, and appears to vary inversely as the tensile strength. As regards tensile strength itself, cast iron has been so much improved in quality in recent years that material possessing a tensile strength of as much as 27 tons per square inch is now obtainable. The range of tensile strength of commercial irons now extends from 15 000 to 60 000 lb. per square inch.

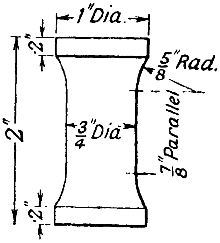


FIG. 57. COMPRESSION TEST PIECE FOR CAST IRON

Tests in direct shear show that the shearing strength of cast iron is somewhat greater than its tensile strength; generally from 10 to 30 per cent, though with irons of low tensile strength the ratio may reach 60 per cent. When tested in torsion a round cast-iron bar fractures along a helix of 45° where the tensile stress is greatest.

Bend Tests. Bend tests on wrought iron and mild steel are carried out in order to determine the ductility. In addition the test provides useful information where the material is to be subjected to cold deformation during further stages of manufacture. Bars are bent double or through a specified angle about a given radius. Reference should be made to various British Standard Specifications. The method of making the bend test is shown in Figs. 71 and 72, pages 100 and 101. The test as usually carried out is qualitative rather than quantitative, but L. W. Schuster has suggested that its usefulness might be extended and that it might, to some extent, replace the tensile test.

CHAPTER V

TESTING MACHINE ACCESSORIES

Necessity for Loading Test Pieces Axially. Accurate results in tensile testing cannot be obtained unless test pieces are properly gripped. It is essential that the line of resultant pull should coincide with the axis of the specimen. The effect of eccentric loading can be gauged from a simple example. Suppose a pull of 10 tons to be applied to a test piece whose section is, say, 1 in. $\times \frac{1}{2}$ in., and that the line of pull is displaced 0·01 in. from the axis towards the short side of the section. In addition to the direct pull there will be a bending moment of $10 \times 0{\cdot}01 = 0{\cdot}1$ ton-in. The stress due to bending is then

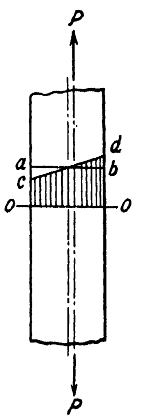


Fig. 58. Stress Variation Over Section of Test Piece Under Eccentric Load

$$f = \frac{M}{Z} = \frac{0{\cdot}1 \times 6}{0{\cdot}5 \times 1^2} = 1{\cdot}2 \text{ tons per in.}^2$$

The stress distribution across the section, instead of being represented by a line *ab*, Fig. 58, whose ordinate corresponds to 10 tons per in.^{2} will be represented by *cd*, the maximum stress being 11·2 tons per in.^{2} and the minimum stress 8·8 tons per in.^{2} As the load is increased the material commences to yield first on the side where the stress is greater, and finally tears from edge to edge. With ductile materials the elastic limit and yield point, calculated as total load divided by area of cross-section, are both lowered if the loading is eccentric, but the ultimate strength appears to be little affected. The ultimate strength of brittle materials is, however, greatly affected by eccentric loading.

Tension Grips. A simple method of holding flat test pieces is by means of a pin at each end. The pins are passed through holes drilled in the test piece, and are supported in shackles in the testing machine.

The more usual method is to employ wedge grips. The grips are simply wedges with serrated faces and are held in a conical hole in the crosshead of the machine. Wedges for flat specimens sometimes have rounded faces as in the Riehle Patent Grip.

Forms of wedges for rounds, flats, and squares are shown in Fig. 59 (*a*), (*b*), (*c*). The wedges should sit fair in the holes in the crossheads and should not project too far. If the wedges

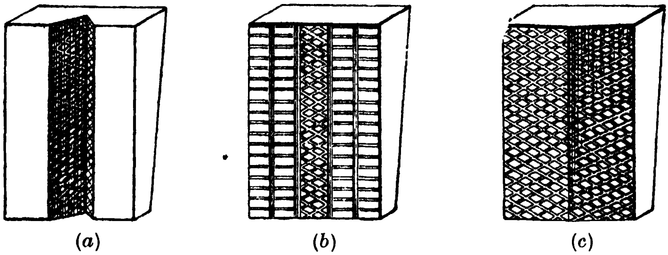


FIG. 59. WEDGE GRIPS FOR FLAT, ROUND, OR SQUARE SPECIMENS
(*Tinius Olsen Testing Machine Co.*)

pull far below the crosshead they are liable to break and the crosshead holes tend to enlarge.

The test piece should be gripped along the whole length of the wedge. Fig. 60 (*a*) and (*b*) shows how the wedges should *not* be arranged. The correct method of gripping is shown in Fig. 61. In Fig. 62 is shown a form of ball and socket liner which aids in securing alignment of the test piece. In some designs, such as the Southwark-Emery, the wedges are made self-adjusting by means of a helical rack and pinion.

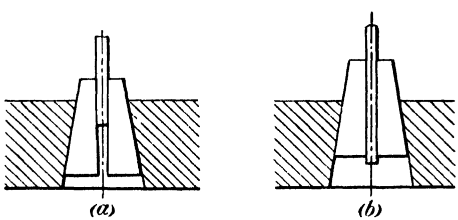


FIG. 60. INCORRECT METHODS OF GRIPPING TEST PIECES

Round specimens are often provided with a head or with screwed ends. Several methods of holding the specimens are in use which aim at securing axial loading. Fig. 63 shows three forms made by Messrs. Olsen. In (*a*) the spherical seat is in the wedge piece. Tool-steel bushes are provided to receive the specimen and either headed or screwed specimens may be accommodated. In (*c*) the holders are provided with a spherical adjustment and are for use with a 0·505 in. diameter headed specimen of aluminium or brass in machines of 10 000 to 20 000 lb. capacity.

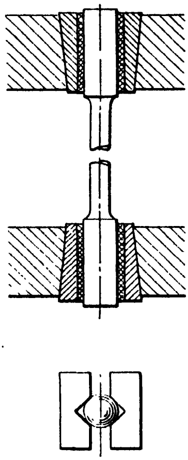


FIG. 61. CORRECT METHOD OF GRIPPING TEST PIECES

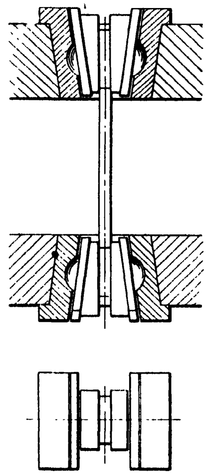
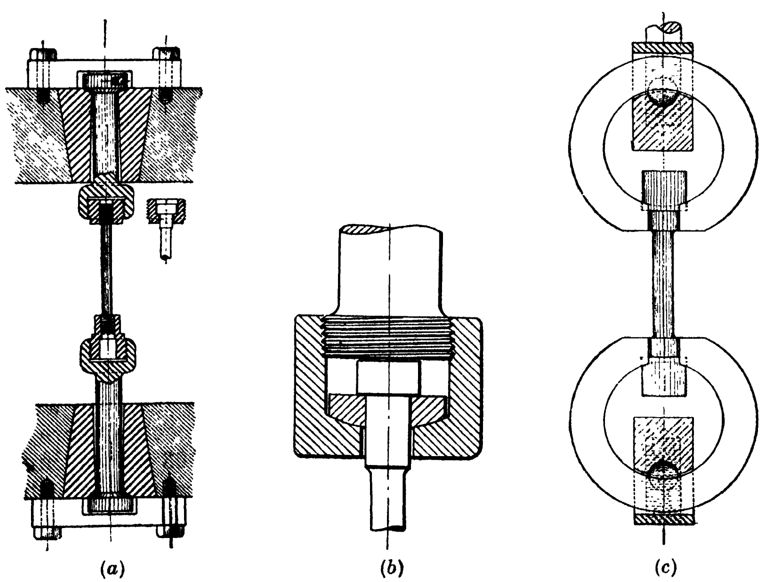


FIG. 62. BALL AND SOCKET WEDGE GRIPS



(a) (b) (c)

FIG. 63. HOLDERS FOR TENSION SPECIMENS WITH SCREWED OR SHOULDERED ENDS

(*Tinius Olsen Testing Machine Co.*)

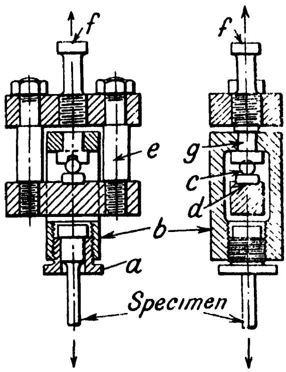


Fig. 64. Robertson's Axial Loading Shackle

A special form of shackle has been devised by Robertson to secure true axial loading, Fig. 64. The specimen is held by screwed split collars *a*, in a socket *b*. The pull is taken on the ball *c* resting on the hard steel seat *d*. The frame *e* is provided with an extension piece *f*, for connection to the grips of the testing machine. The hole for the ball socket *g* and the threaded hole for the split collar should be machined at the same setting in order to secure alignment.

Fig. 65 shows a grip for specimens of thin sheet metal. Another form is shown in Fig. 66. The shackle consists of two steel plates supported on a pin in the head *b* which is screwed into the holder in the pulling head of the machine.

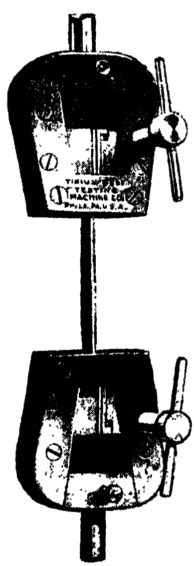


Fig. 65. Wedge Grips for Sheet Metal Test Pieces
(Tinius Olsen Testing Machine Co.)

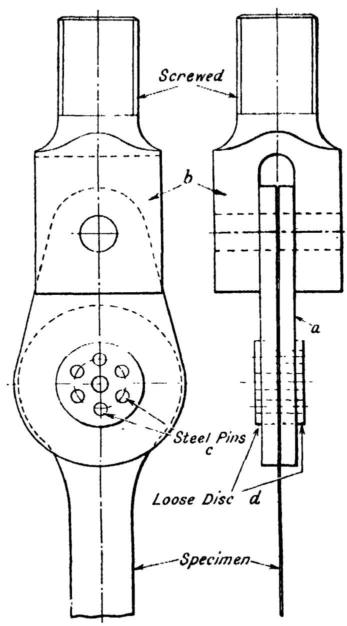


Fig. 66. Shackles for Sheet Metal Test Pieces

The plates are drilled to receive two loose discs *d*. The disc and the sheet metal specimen are drilled to receive steel pins *c* which transmit the pull.

For testing strip of soft material which is liable to tear at the pin holes a device similar to that shown in Fig. 69 may be used with advantage. A form of holder widely used in U.S.A. is the Templin grip manufactured by R. L Templin of the Aluminium Company of America.

A serviceable method of gripping wires consists in attaching the vice jaws to a holder in the machine head by a lazy tongs linkage. The design is such that as the load is applied the pressure between the jaws and the wire tends to become concentrated towards the rear, thus relieving the pressure at the point where the wire enters the jaws.

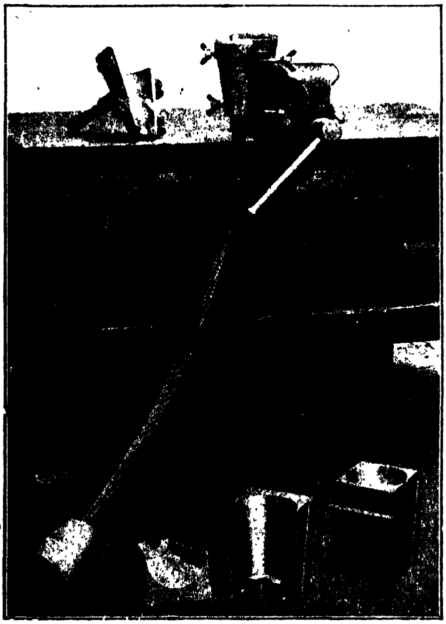


FIG. 67. METHOD OF CAPPING WIRE ROPES

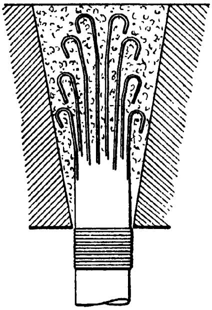


FIG. 68. SECTION THROUGH CAP

Testing of Ropes and Chains. Ropes are somewhat troublesome to test owing to possible damage to the rope at the grip. One method consists in casting a conical head at each end of the test piece. The ends of the rope are unwound for several inches depending on the size of the piece, being first bound up to prevent further separation of the wires along the rope.

The wires are then splayed out, cleaned, and tinned. Next, the ends of the wires are bent over and. using a conical mould, are encased in an alloy of low melting point. (Figs. 67 and 68.) An alloy of 83 per cent lead and 17 per cent antimony is sometimes used. Whatever metal is employed, it is important

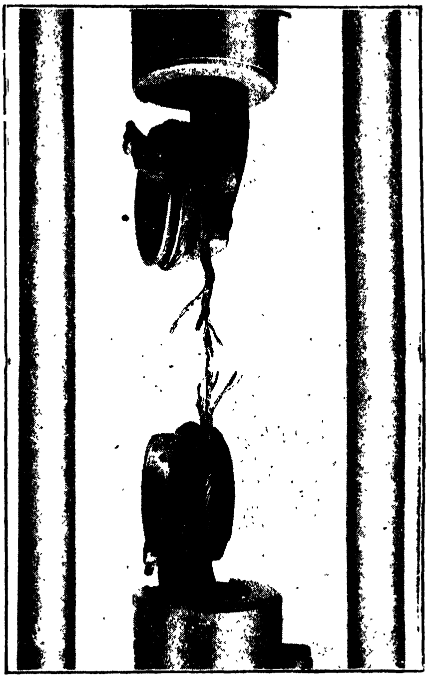


FIG. 69. METHOD OF HOLDING THIN ROPES
(*Alfred J. Amsler & Co.*)

not to overheat the wires. Split conical grips are used to hold the rope in the testing machine. The foregoing method is tedious, and to obtain a successful test care must be exercised in making the cap. A suitable alloy is now specified by the B.S.I.

Long, lightly serrated wedges are sometimes used. They should be relieved at the point where the rope enters the grip. Another method is to use a series of wedge grips one behind another. In the Amsler system the wedges are lined with thin strips of wood to prevent damage to the rope. When under test the cable is forced into the wooden liners and suffers no injury.

For thin ropes special drums are used. This, a simple and effective method, is illustrated in Fig. 69.

Yet another method is to employ double wedges embodying the principle of the Reliance Rope Grip, Fig. 70. The centre

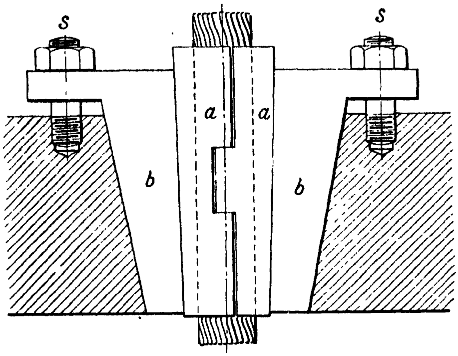


FIG. 70. RELIANCE ROPE GRIP

wedges *aa* are of fairly soft steel and are driven in as testing proceeds by means of a sledge. The harder strands of the rope

FIG. 71. COLD BEND TEST. FIRST STAGE
Bending the specimen by means of a rounded punch.
(*Alfred J. Amsler & Co.*)

embed themselves in the softer material of the wedge and an effective grip is obtained. There is a strong tendency to force the wedges *bb* out of their socket and it is advisable to anchor them by studs *ss* to prevent slacking back.

Chains may be gripped by passing steel pieces, bent into U

form, through each of the end links. The open ends of the U's are then gripped in the same manner as flat bars. Messrs. Olsen supply special blocks by means of which separate lengths of the chain may be subjected to the proof load.

Bending, Shear and Compression Tools. In making bend

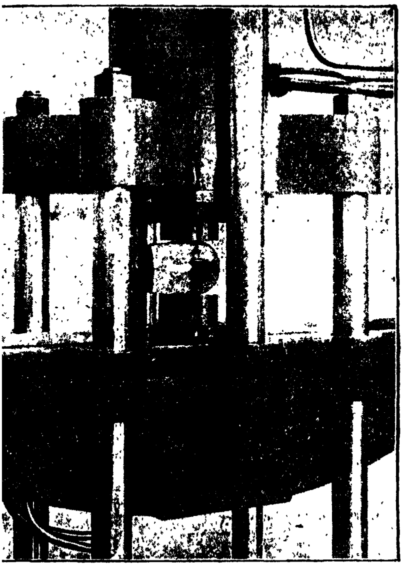


FIG. 72. COLD BEND TEST. FINAL STAGE
(*Alfred J. Amsler & Co.*)

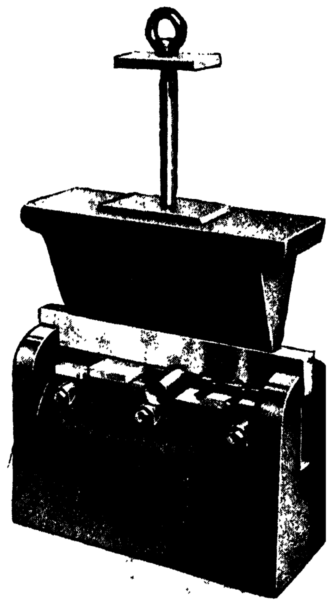


FIG. 73. OLSEN SHEARING TOOL
(*Edward G. Herbert Ltd.*)

tests, the test bar is supported on roller bolsters firmly clamped to the machine table. When a bar has to be bent double it is first bent as in Fig. 71, using a punch of specified radius, and finally closed as shown in Fig. 72.

Specimens cut from plates should be planed on their edges and the corners slightly rounded to avoid the formation of small cracks as the specimen is bent.

In making shearing tests the specimen, round or square, is passed through a system of three rings with sharp edges, the middle ring being allowed to slide between the other two. The test piece must fit the bore of the rings exactly in order to minimize the bending action.

Another form of shearing tool is shown in Fig. 73. It can be used for plates up to $\frac{1}{2}$ in. thick by 5 in. long and rounds of $\frac{1}{2}$ in., $\frac{3}{4}$ in. and 1 in. diameter. The maximum capacity is 200 000 lb.

Spherical seats are necessary for making accurate compression tests. The axis of the seat must be in line with the axis of

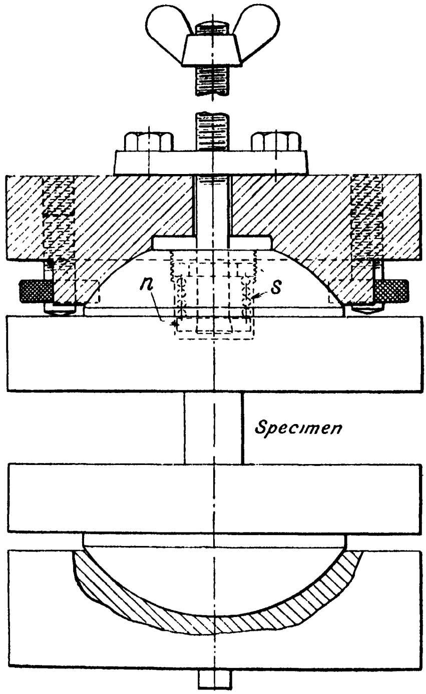


FIG. 74. COMPRESSION PLATES WITH SPHERICAL SEATS
(*A. Macklow Smith*)

the testing machine and the centre of curvature of the spherical surface should be on the face of the platen.

One form of spherical compression plate for use in a vertical machine is shown in Fig. 74. The upper seat is bored and screwed to receive the head of a bolt which prevents the seat from falling out. To permit lateral freedom the seat is supported on the spring *s* by the nut *n*. It is customary to fit a hardened steel insert in the centre of the compression plate to obviate damage to the surface when testing pieces of hard metal in compression.

CHAPTER VI

EXTENSOMETERS AND RECORDERS

Measurement of Extension. The extension occurring within the elastic range in a material under stress is, with the short specimen generally adopted for tensile tests, of comparatively small magnitude. For instance, a test piece having a gauge length of 2 in. is extended only by about 0·002 in. when the elastic limit is reached. Considerable refinement of measurement is needed to measure such small extensions with accuracy.

Instruments specially constructed for the measurement of strains are termed extensometers. Their operation may be based on—

(1) The use of a micrometer or vernier;
(2) Indication by means of a dial;
(3) Reading the extension by means of a microscope;
(4) Mechanical magnification;
(5) Optical magnification;

or a combination of these methods.

For measuring extensions after fracture, a vernier calliper may be used, but for measurement of elastic extension this instrument is not sufficiently refined.

To obtain accurate results when measuring the elastic extension it is advisable to take measurements on opposite sides of the test bar, as otherwise initial curvature of the bar may be the cause of serious errors. With some instruments separate sets of readings have to be taken; in others the resultant of the strains on opposite sides of the specimen is averaged by the instrument itself.

When finding the modulus of elasticity (E) from sets of readings taken on opposite sides of the test piece the value should be calculated from the mean of the two values of the strain. Otherwise, if E_1 and E_2 be the values of the modulus of elasticity obtained for the respective sides of the bar the mean value of E is given by

$$E = 2E_1E_2/(E_1 + E_2)$$

Many forms of extensometer have been devised. The following descriptions of the construction and application of the more important commercial types should enable the mode of operation of any instrument to be readily grasped.

Olsen's Duplex Micrometer Extensometer. This instrument consists of two frames which grip the test piece at the gauge marks by steel points and two knife-edges, Fig. 75. The lower frame carries two micrometers, the points of which register with two steel plugs carried by the upper frame. As extension takes place the micrometer screws are turned until their points make contact with the respective faces of the plugs. Contact is indicated by the ringing of an electric bell operated by a weak electric current, or, better still, by the aid of a telephone receiver.

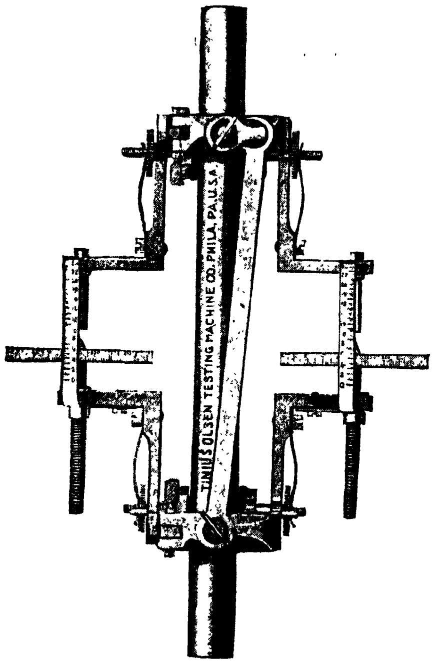


FIG. 75. OLSEN DUPLEX MICROMETER EXTENSOMETER
(*Edward G. Herbert Ltd.*)

Extensions are observed from the graduations on the vertical scales and the micrometer heads.

The instrument may be used on round, flat or square specimens and is easily adjusted to the size of the piece. Readings may be taken to within 0·0001 in. To assist in setting up the instrument the frames are connected by links which are removed when the instrument is properly attached to the specimen.

Instruments of this type require careful use in order to prevent pressure of the fingers from disturbing the setting of the instrument. In another form, the micrometer screws are replaced by indicating dials.

Unwin's Extensometer. Unwin's extensometer consists of

two frames each carrying a small spirit level. It is shown diagrammatically in Fig. 76. The two frames are clamped to the test piece by means of pointed set screws *S*, *S*, the lower frame being supported by the adjusting screw *C* which serves to level the lower frame. The upper frame is supported on a point of the micrometer screw. The lower frame is first levelled and then the upper. The micrometer reading is noted and after applying a load the upper frame is again levelled by the micrometer screw *M*. The difference between the two readings gives the extension of the specimen over the gauge length. The micrometer reads to 0·0001 in.

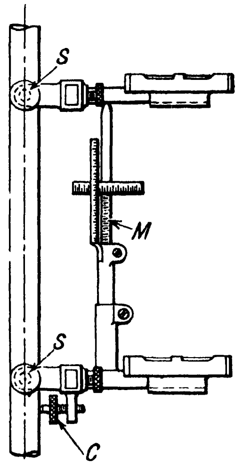


FIG. 76. UNWIN'S EXTENSOMETER

The Huggenberger Extensometer. The Huggenberger extensometer is of the mechanical type and registers extensions by means of a system of levers. It is shown diagrammatically in Fig. 77 and consists of a fixed knife-edge *A* attached to the frame of the instrument, and a double knife-edge *B*. The double knife-edge is free to tilt about the point *C* on the frame. A lever *BD* extends from the knife-edge and is coupled at *E* to a pointer pivoted at *G*. The pointer moves over a scale from which the extension of the specimen can be observed. A mirror behind the pointer assists in avoiding parallax.

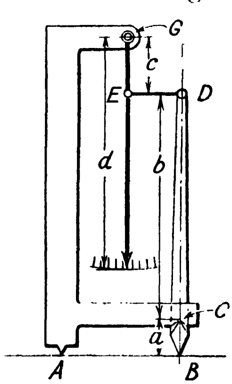


FIG. 77. HUGGENBERGER EXTENSOMETER (DIAGRAMMATIC)

Several forms of this extensometer are made, of varying degrees of sensitivity. The most sensitive type is intended for laboratory purposes while the more robust and less sensitive types are for use under workshop conditions.

Referring again to Fig. 77, an extension e between the points A and B is increased to $e \times [(b \times d)/(a \times c)]$ on the scale. In the most sensitive type the magnification is 1 200 and in the other types, 1 000 and 300 respectively. The instrument is about 160 mm. high, 15 mm. deep and

weighs about 70 g. There are 38 scale divisions each of 0·05 in. One division in the most sensitive instrument corresponds to a change of length between 1 in. gauge points of 0·000042 in. The total 38 divisions correspond to 0·0016 in.

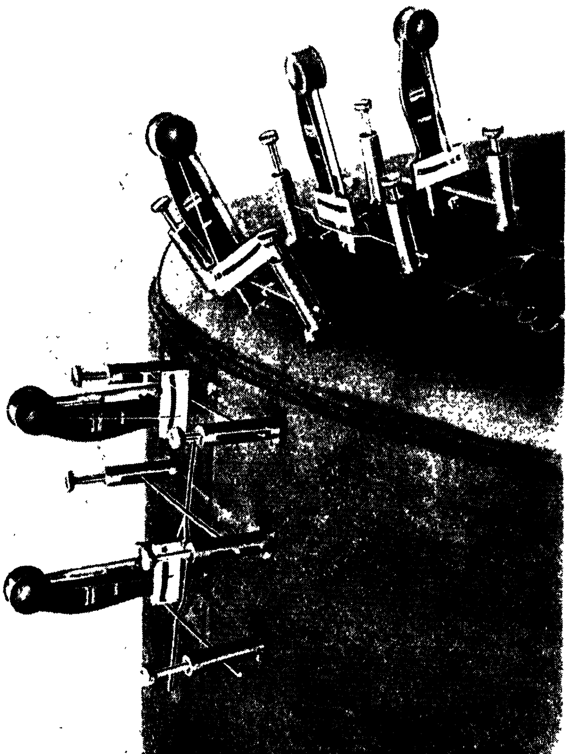


FIG. 78. HUGGENBERGER EXTENSOMETERS CLAMPED TO SURFACE OF A PRESSURE VESSEL
(*Howden & Co.*)

The standard gauge length is 1 in. and this can be changed to ½ in. by reversing the position of the stationary knife-edge. The gauge length should always be checked after this removal. For greater gauge lengths of 2 in. or 4 in. extension bars are provided. In determining Young's modulus it is necessary to use two instruments, one on each side of the test bar.

If N is the lever multiplication,
L the gauge length in inches,
Z the actual scale reading,
X the extension in inches per inch of gauge length,
S the stress in lb. per in.^{2}
E Young's modulus,

then $X = Z/(N \times L)$; $S = (E \times Z)/(N \times L)$;
$E = (S \times N \times L)/Z$.

The Huggenberger extensometer is specially adapted for the measurement of the small strains that occur in loaded structures

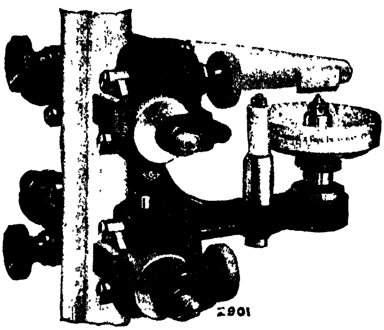


FIG. 79. CAMBRIDGE EXTENSOMETER
(*Cambridge Instrument Co. Ltd.*)

and machine parts. Fig. 78 shows a number of these extensometers clamped to the external surface of a pressure vessel.

To set the instrument it is first clamped to the specimen with the pointer in the middle of the scale and the locking device closed. The locking device is then opened and the specimen subjected to stress in order to ascertain the direction and amplitude of the pointer. If the pointer swings beyond the scale it must be reset. In the less sensitive types a resetting device is provided. Otherwise, the instrument must be tilted on the moving knife-edge until the stationary knife-edge is disengaged and the extensometer displaced in its plane until the pointer comes to the desired position. The instrument is then gently tilted back and reclamped.

To ascertain if the fixing is correct, the narrow edge of the scale is touched with one finger, causing the pointer to move. If the pointer returns to its initial position when the finger is removed it may be assumed that the initial setting is correct.

The Cambridge Extensometer. The Cambridge Extensometer, Fig. 79, combines mechanical magnification with a micrometer, and is designed for use as a workshop instrument. It consists of two separate portions each of which is attached to the specimen by means of steel points. The points are formed on the ends of hard steel rods which work in a geometric slide. After the points are centred they can be clamped in position by means of knurled nuts. The two portions are pivoted together upon a steel knife-edge. From the upper portion extends a flexible nickel steel tongue which acts as a lever to magnify the extension of the specimen. Movement of the

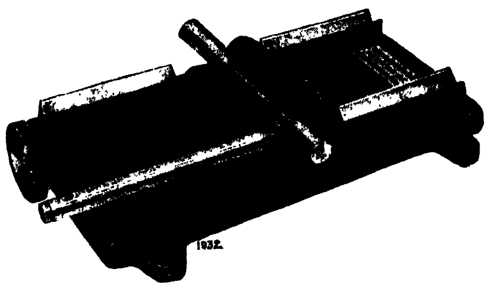


FIG. 80. MARKING-OFF TOOL FOR 2 IN. GAUGE LENGTH
(*Cambridge Instrument Co. Ltd.*)

tongue towards or away from the point of the micrometer screw, carried in the lower frame and clearly seen in the illustration, is five times the actual extension of the specimen.

To use the instrument the steel tongue is set vibrating and the micrometer screw turned until its point just touches the tongue as this vibrates. It is important not to cause the tongue to deflect too much as a false reading is then liable to result. A slight vibration is all that is necessary.

The standard instrument is supplied for use on specimens up to $\frac{3}{4}$ in. diameter and for gauge lengths of 2 in. or 4 in.

Readings may be estimated to 1/20 000 in.

A marking-off tool, Fig. 80, is supplied for marking the test pieces accurately. It consists of a cast iron base having two parallel V-grooves separated by the gauge distance it is desired to employ. In a gap in the middle of the grooves, and set at right angles to them, is a third groove in which the test piece is clamped. A steel centre-punch is placed in turn in each longitudinal groove and by tapping the punch with a hammer the gauge points may be marked in the correct positions.

The Ewing Extensometer. Ewing's extensometer enables the variation in length of the specimen under test to be continuously watched. It can be used either in the vertical or the

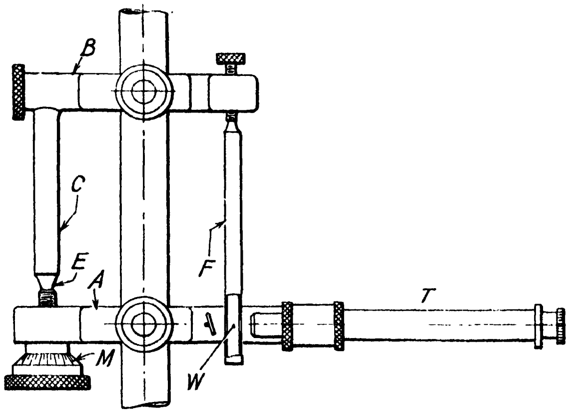


FIG. 81. EWING EXTENSOMETER (DIAGRAMMATIC)

horizontal position. The instrument is illustrated in Figs. 81 and 82. Referring to Fig. 81, *A* and *B* are two clamps pinched on to the test piece by pointed set screws. The clamp *B* carries

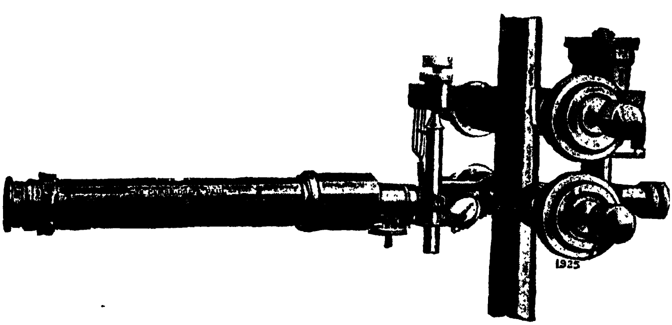


FIG. 82. EWING EXTENSOMETER
(Cambridge Instrument Co. Ltd.)

a rod *C* having a ball end *E* which rests in a seat in the micrometer screw *M* in clamp *A*.

The light tube *F* is held in contact with the adjusting screw *K* in the clamp *B* on the other side of the test piece by means of

a spring. This is clearly seen in Fig. 82. The tube is pierced diametrically and carries a fine wire W which is viewed by the aid of the microscope T. The readings are observed on the micrometer scale in the eyepiece of the microscope. The micrometer screw has 50 threads to the inch and one complete turn corresponds to 50 divisions on the eyepiece scale. When the test piece elongates, the displacement of the wire is twice that of the gauge length on account of the leverage about the fulcrum E. As a result one division on the scale is equivalent to 0·0002 in. extension on the gauge length. Readings to 0·00002 in. are possible. The standard instruments are for gauge lengths of 8 in. and 2 in.

A modified form is made for measurement of the elastic compression of short blocks on a 2 in. gauge length. Readings can be taken by estimation to 0·002 mm. or 0·000008 in. A clamping bar is provided for registering the clamps on the test piece.

Mirror Extensometers. Mirror extensometers operate on the following principle.

Consider a scale, telescope and mirror set up as indicated in Fig. 83, with the mirror in the position *aa*. Let *oo′* be the line

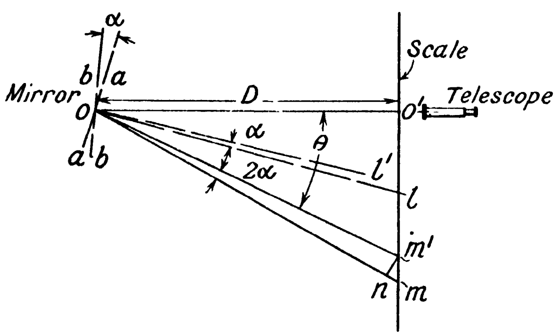


FIG. 83. PRINCIPLE OF MIRROR EXTENSOMETER

of sight and *ol* the normal to the mirror. The scale division seen in coincidence with the cross wire of the telescope is *m*, say, the angle of incidence *mol* being equal to the angle of reflection *loo′*. If the mirror be tilted to the position *bb*, the normal will be *ol′* and the scale division in coincidence with the cross wire of the telescope is *m′* where $\angle\, m'ol' = \angle\, l'oo'$. If α is the angle through which the mirror is tilted this will also be the

angle between the normals, i.e. the angle lol'. It is easily seen that the angle mom' between the two positions of the incident ray is equal to (twice $\angle loo'$ — twice $\angle l'oo'$) $= 2\alpha$.

$$\text{Now} \quad mm' = o'm - o'm'$$
$$= D(\tan \angle moo' - \tan \angle m'oo')$$
$$= D(1 + \tan \angle moo' \times \tan \angle m'oo') \tan (\angle moo' - \angle m'oo')$$
$$= D(1 + \tan \angle moo' \times \tan \angle m'oo') \tan \angle mom'.$$

where D is the distance between the scale and the mirror.

If the angle moo' is small the angle 2α will also be small, and we can neglect the product of the tangents and in place of tan 2α we may write $\tan 2\alpha = 2 \tan \alpha = 2 \sin \alpha = 2\alpha$.

If the angle moo' is of the order of $5°$, $\tan \angle moo'$ is about 0·1 and the product of the tangents is about 0·01, so the error introduced by this approximation is about 1 per cent.

If mm' corresponds to 1 scale division we have $\alpha = 1/2D$ as the value of the angle of tilt in radians corresponding to one scale division. The distance D between the scale and the mirror must be measured in the same units as the divisions of the scale.

When, as is sometimes the case, a circular scale of radius D is used the foregoing approximation becomes exact.

Another approximation which is sometimes useful when the working angle increases beyond that in which the tangent can be taken as the angle itself is the following—

From the figure, we have

$$\tfrac{1}{2} \angle m'om = \tfrac{1}{2}(nm'/om') = \tfrac{1}{2}mm'(\cos\theta/D\sec\theta)$$
$$= \cos^2\theta/2D$$

as the value of one scale division in radians.

Bauschinger's Extensometer. Bauschinger introduced a type of instrument in which two clips were pressed against the specimen by springs pressing outwards against caoutchouc rollers. The rollers were carried by a clip and themselves carried the mirrors. Extension of the specimen caused rotation of the mirrors and the rotations were measured by means of scales and telescopes. Results were measurable to 0·0001 mm. A modern form employing steel rollers will be described later.

Morrow's Extensometer. Morrow's instrument is shown in Fig. 84. Two rings R, R, are clamped to the specimen by set

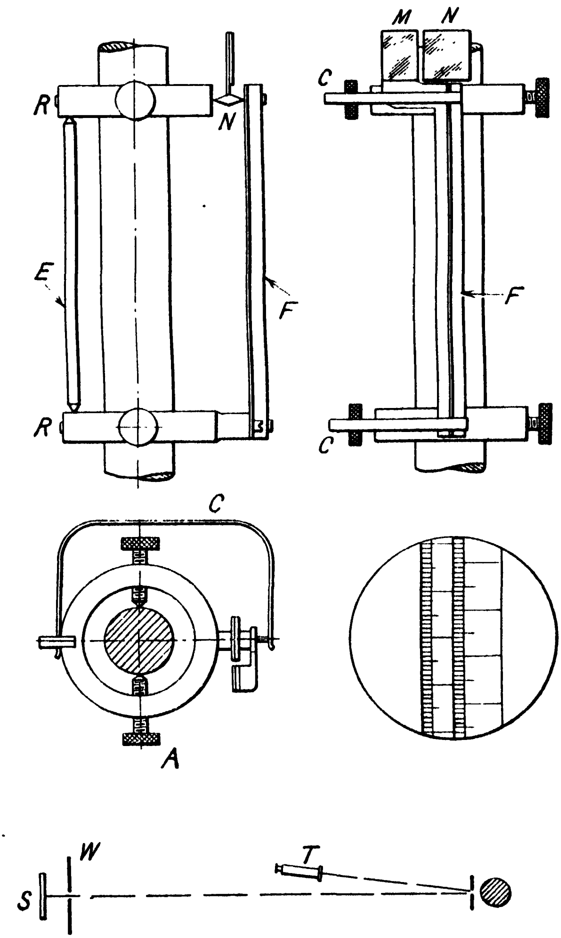


FIG. 84. MORROW'S EXTENSOMETER

screws A round which they pivot. The rod E allows the rings, in addition, to pivot about its ends.

An arm F rigidly attached to the lower ring carries a fixed mirror M at its upper end and supports a rhomb lever

which carries the mirror N. The apparatus is held in position by clips C.

A scale S placed parallel with the axis of the test piece is illuminated by a lamp, not shown, and the reflected beam allowed to pass through an aperture in the screen W to the extensometer mirrors and thence by reflection to the telescope T.

The extension of the specimen causes the rings to pivot round the ends of the rod E, thus inducing relative motion between the upper ring and the arm F. This tilts the rhomb and its mirror and the angle of tilt is observed through the telescope.

The appearance of the scale in the telescope is then as shown in the middle right-hand view in the figure, the image of the scale shown by the tilting mirror appearing displaced from that shown by the fixed mirror.

With a scale divided into fortieths of an inch and with a scale set 80 in. from the mirror, the magnification is 3 000 : 1 so that the smallest reading is

$$\frac{1}{40} \times \frac{1}{10} \times \frac{1}{3\ 000} = \frac{1}{1\ 200\ 000}$$

of an inch.

Lamb's Roller Extensometer. The roller extensometer, Fig. 85, consists essentially of two elements held together against opposite sides of the test piece by means of light springs. Each element is complete in itself and consists of a pair of hardened steel plates separated by two small rollers which are held in position by the springs. Each plate has a sharp knife-edge which bears on the face of the test piece. As the test piece elongates the plates move relatively to each other and give the roller a small rotary movement proportional to the extension of the specimen.

Each roller carries a mirror at one end and at the other a small knurled head for making the final adjustment. No jig for marking the specimen is needed. A pair of dowel pins or a gauge is used to set each element to length before it is clamped to the specimen, the dowels being withdrawn before straining the test piece. The theory of the instrument, as given by Professor Lamb, is as follows—

Referring to Fig. 86, d_1 and d_2 are two rollers—one on each side of the test piece—each carrying a mirror. Let D be the

distance between the scale and mirror and a the distance between the two mirrors. Let θ and ϕ be the incidence and

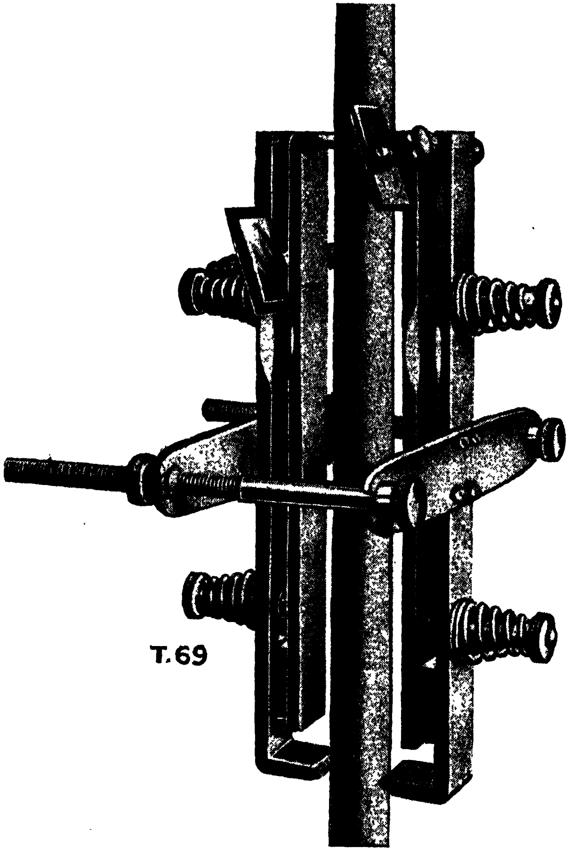


FIG. 85. LAMB'S ROLLER EXTENSOMETER
(*A. Macklow Smith*)

reflection angles at the mirrors A and B and let s be the reading on the scale.

Then, for small opposite rotations $\delta\theta$ and $\delta\phi$ of the mirrors the increment of scale reading is

$$\begin{aligned}\delta s &= 2D(\delta\theta + \delta\phi) + (2a\delta\theta/\cos\phi)\cos\phi \\ &= 2D(\delta\theta + \delta\phi) + 2a\delta\theta \quad . \quad . \quad . \quad . \quad (1)\end{aligned}$$

Again, if e_1 and e_2 are deformations given to each extensometer element, d_1 and d_2 the corresponding roller diameters, we have the relations

$$\left.\begin{aligned} e_1 &= d_1\delta\theta \\ e_2 &= d_2\delta\phi \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \qquad (2)$$

and the axial deformation is

$$e = \tfrac{1}{2}(e_1 + e_2) = \tfrac{1}{2}(d_1\delta\theta + d_2\delta\phi) \, . \quad . \quad . \qquad (3)$$

Now any effect of non-axial loading of the specimen will be

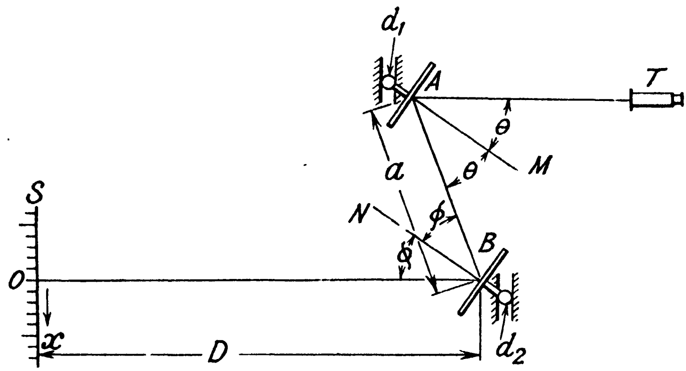


FIG. 86. PRINCIPLE OF LAMB'S EXTENSOMETER

to make a small proportional difference in the values of e_1 and e_2 and we can therefore write

$$e_2 = (1 + k)e_1 = (1 + k)d_1\delta\theta \quad . \quad . \quad . \qquad (4)$$

and if there is a small difference in the roller diameters we can (writing d in place of d_1) put

$$d_2 = (1 + \lambda)d \quad . \quad . \quad . \quad . \qquad (5)$$

Hence, from equations (2), (4) and (5)

$$(1 + \lambda)\delta\phi = (1 + k)\delta\theta \quad . \quad . \quad . \qquad (6)$$

From (1) and (6),

$$\delta s = \left[2D\,\frac{2 + k + \lambda}{1 + \lambda} + 2\alpha\right]\delta\theta \quad . \quad . \qquad (7)$$

and from (3), (5) and (6)

$$e = \tfrac{1}{2}d(2 + k)\delta\theta \, . \quad . \quad . \quad . \qquad (8)$$

Combining (7) and (8)

$$e = \frac{d\left(1 + \frac{k}{2}\right)\delta s}{2D\,\frac{2 + k + \lambda}{1 + \lambda} + 2a} \quad . \quad . \quad . \quad (9)$$

which, omitting squares and products of small quantities reduces to

$$e = \frac{d(1 + \lambda/2)\delta s}{4(D + a/2)} \quad . \quad . \quad . \quad . \quad (10)$$

In this result it is to be noted that $d(1 + \lambda/2)$ is the mean of the diameters of the two rollers.

The measurements upon which the accuracy of the performance of the instrument depends are—

(1) diameter of the rollers,
(2) gauge length of the instrument,
(3) the scale distance,
(4) distance between mirrors,
(5) the reading of the scale.

Of these (1) and (2) constitute the permanent calibration of the instrument, the degree of accuracy to which the former is known being of the order of 0·005 per cent to 0·02 per cent according to the size of roller employed, while the latter is within 1 per cent of the nominal length. The magnitude of the errors arising from (3), (4) and (5) depends upon the operator's choice of a suitable distance for the scale in relation to the amount of extension measured. With reasonable care (3) and (4) can be measured easily to an accuracy of 0·2 per cent for any distance of the scale from the instrument and as it is usual to arrange on the average for scale deflections of the order of 30 cm., there is no difficulty in securing an accuracy in the case of (5) of 0·15 per cent. It will be seen, therefore, that the worst possible combination of errors enumerated cannot lead to an error in the result exceeding 0·5 per cent and with the ordinary precautions usual in work of this nature an accuracy of 0·2 per cent is easily obtainable.

Instead of using a scale and telescope, a projector lamp, Fig. 87, may be employed. The projector is set up as near to the mirror as possible, the mirrors are adjusted and the lamp focused until the crosswires are clearly defined.

Observations may be plotted directly on squared paper on a rotating drum.

When a telescope is used it is convenient to employ a millimeter scale at a distance of 5 000 $(d/4)$, where d is the mean diameter of the rollers. In this case 1 cm. represents 0·0002 in. At such distances it is easy to read to $\frac{1}{2}$ mm., or an extension of

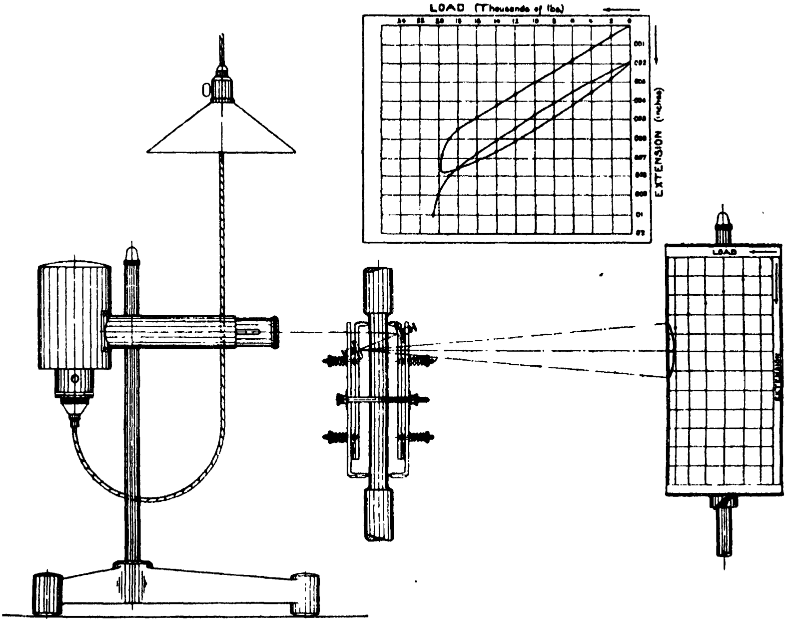


FIG. 87. METHOD OF PLOTTING LOAD-EXTENSION CURVE WITH THE AID OF A PROJECTOR LAMP
(*A. Macklow Smith*)

0·00001 in. Lamb's Extensometer is made by Mr. A. Macklow Smith, Westminster.

The Amsler Mirror Extensometer. Amsler's mirror extensometer is based on the Martens' system and is shown in principle in Fig. 88. In this instrument comparison is made of the amount of the extension of the test piece relative to a standard length in the form of a strip. The *comparison strip*, as it is termed, has at one end, a sharp edge which is placed in one of the gauge marks on the specimen. At the other end of the strip is a notch which carries a little lever supported in the other

gauge mark. The lever, in the form of a rhomb, has sharp parallel edges and carries a small mirror. When the test piece undergoes alteration in length the rhomb is tilted. An elastic clip holds the comparison strip against the specimen and prevents the point of contact of the small lever from being displaced on the test piece without, however, interfering with the angular movement of the lever. The change of gauge length is

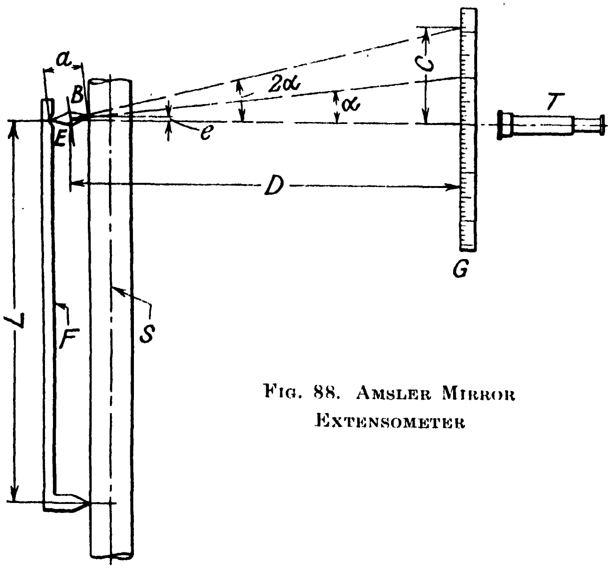


FIG. 88. AMSLER MIRROR EXTENSOMETER

measured by noting the angular movement of the mirror by means of a telescope and scale. Referring to Fig. 88, *T* is the telescope, *B* the mirror, *E* the rhomb or prism lever, *S* the test piece, *F* the comparison strip, and *G* the scale. A similar assembly is arranged symmetrically on the opposite side of the test bar. The illustration shows only one set of measuring parts, and for greater clearness the clip is not shown.

For reasons of stability it is usually preferable to place the edge of the strip *F* at upper end of the specimen and the mirror carrier at the bottom.

In the diagram, *a* represents the width of the prism, *D* the distance between scale and mirror, *L* the length of the comparison strip measured from the knife-edge to the middle of the

notch and consequently the initial length on which the extension is to be found.

Let it be assumed that in the original position of the test bar the scale and mirror are parallel and that the telescope is at the same height as the mirror and perpendicular to it.

If the test bar be deformed by a slight elongation e, the mirror will turn through the angle α with respect to its original position and the relation between the angle and the extension is

$$a \sin \alpha = e$$

By taking a reading with the telescope, a point on the scale is seen which is distant c from the original reading. Since the angle of reflection is equal to the angle of incidence the original ray is turned through an angle 2α and we have the relation

$$c = D \tan 2\alpha$$

As α is usually small we may assume $\tan 2\alpha = 2 \sin \alpha$. Substituting this value in the second equation and dividing one by the other we obtain

$$c = 2(De/a)$$

or

$$e = (ac/2D)$$

showing that e is proportional to c.

If the angle is too large to allow of this approximation being made, the exact value of e must be found as a function of c by means of the first two equations.

To determine the mean change in length it is necessary to take measurements on the two sides of the bar. This necessitates two telescopes and two scales. They are conveniently mounted on a cross bar attached to a vertical rod and can be used equally well with a horizontal or vertical type of testing machine.

The distance between the scale and the mirror may be conveniently measured by means of a wooden lath cut to any desired length. The width of the rhomb knife-edge is 4 or 5 mm. according to the length of the comparison strip.

The distance between the scale and mirror should be selected so that a magnification of 500 is attained, i.e. $c = 500\, e$. This gives $2D/a = 500$ or $D = 250a$.

With a magnification of 500 a tenth of a millimetre on the scale corresponds to 1/5 000 mm. change of length of the test bar.

With the previous notation, if S is the sum of the two readings then

$$e/S = a/4D$$

The position of the knife-edge rhomb should be as nearly perpendicular as possible to the surface of the test bar. To facilitate the adjustment of the mirror carrier to the perpendicular position each carrier is provided with an index at right

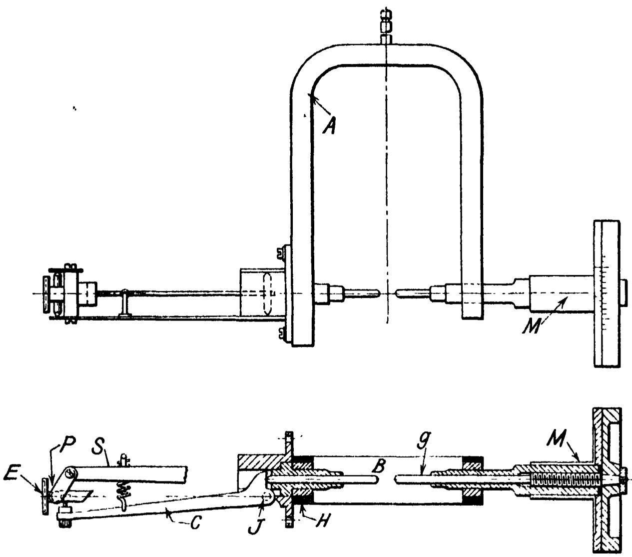


FIG. 89. COKER'S LATERAL EXTENSOMETER

angles to the knife-edge rhomb; its position ought therefore to be parallel to the surface of the test bar for the correct position of the knife.

If desired, a mirror may be attached to the comparison strip alongside the mirror carried by the rhomb. This enables the zero on the scale to be checked as the test proceeds and any bodily shift of the test bar to be allowed for if necessary.

The Gerard Extensometer. The Gerard Extensometer is a compact, sensitive, and robust instrument which indicates the extension on a 2-in. gauge length by means of a pointer moving round a circular dial. The instrument is shown diagrammatically in Fig. 90.

One of the features of its design is that the instrument is constructed so that all the forces act in one plane. The mechanism comprises the frame and fixed parts *A*, and a moving clamp and member *M* which transmits the extension of the test

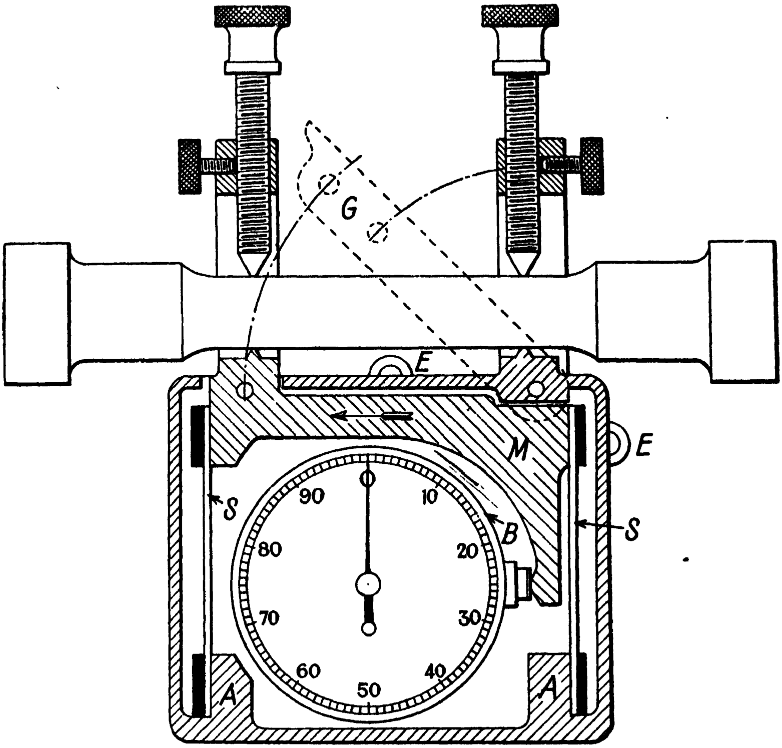


FIG. 90. DIAGRAMMATIC REPRESENTATION OF GERARD EXTENSOMETER

piece to the dial indicator. The parallel motion of the member *M* is effected by the flat springs *S*.

Each scale division is 0·05 in. wide and represents an extension of 0·0001 in., or an extension of 0·005 per cent of the 2-in. gauge length. Otherwise, one revolution of the pointer over a scale 5 in. long represents an extension of 0·01 inch. The maximum movement is about 0·05 in. or 2·5 per cent of the gauge length, which is well into the region of plastic extension of metals.

There is no relative rotation of the gauge points on the test piece so that ball points may be substituted for the usual sharp conical points, an advantage when testing very hard or thin material.

The distance between the gauge points may be locked at the gauge length before clamping the extensometer on to the test piece, thus obviating prior marking of the specimen. The locking gate *G* also locks the adjustable bezel *B* of the indicator, and in addition provides means for preventing the extensometer from falling off the test piece.

The instrument can be used on strip up to $1\frac{1}{8}$ in. wide and on round bar up to 1·128 in. diameter. Special V grip attachments are provided for use on wire.

A certain amount of weight is essential to meet the requirements of quick commercial work without derangement, but the effect of this weight on light test pieces can be neutralized by suspending the instrument with a very flexible spring attached to one of the eyelets *E*, according as the instrument is used in the vertical or horizontal position.

Extensometer for Measuring Lateral Strains. Coker's lateral extensometer is shown in Fig. 89. The U-shaped frame *A* carries a micrometer *M* in a friction-tight grip and is provided with a loose measuring needle. A second similar needle *B*, in line with the first is supported in a casing *H*, and its outer end bears against the short arm of a bell-crank lever *C*, pivoted at *J*. The longer arm of the bell crank is kept in contact by a light spring with the projecting arm *P* of the mirror pivoted to a member *S* of the frame *A*. Any displacement between the tips of the measuring needles causes a tilt of the mirror and serves to measure the lateral strain produced in a stressed bar placed between them.

The extensometer frame is held in a convenient support when applied to measure the change of dimension of a specimen. The mirror is employed in conjunction with a lamp or telescope and scale in the usual way.

The Tuckerman Optical Strain Gauge. The Tuckerman Optical Strain Gauge made by the American Instrument Co. is suited for the measurement of tension and compression strains as small as 0·000002 in.

The instrument consists of an auto-collimator and an extensometer or gauge (Fig. 91). The gauge element carries a small prism and makes contact with the test piece through

a knife edge at one gauge point and through a lozenge at the other (Fig. 92). The strain is made manifest by the rocking of the lozenge which forms the mirror element.

The principle of the instrument will be clear from the diagram (Fig. 93), which shows the reticule plane of the auto-collimator BFC, the objective lens L, and the lozenge face and roof prism represented as plane mirrors M_1, M_2, inclined at an angle X. The point F on the reticule represents the fiduciary mark which is illuminated by a small lamp placed behind it. The reticule lies exactly in the focal plane of the objective lens L which is highly corrected for aberration. Light from F passes through lens L and emerges as parallel light. The light striking M_1 is reflected to M_2 and thence through L to form an image of F at C. Light reflected by M_2 follows a similar but reversed path to form an image of F at B. Since section A of the reticule is the only portion visible through the Ramsden eye-piece, B is the image which is seen.

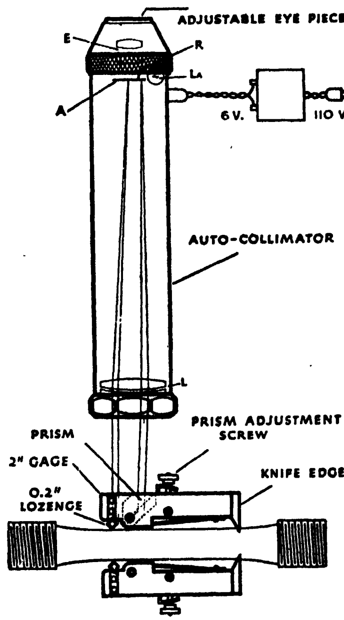


FIG. 91. RELATIONSHIP OF THE AUTO-COLLIMATOR AND EXTENSOMETERS, AND THE RELATIVE POSITIONS OF THE OPTICAL COMPONENTS OF THE AUTO-COLLIMATOR

Rotation of M_1 relative to M_2 causes a displacement of the image B relative to F. The single light beam shown in Fig. 91 corresponds to one of those shown in Fig. 93. The fiduciary mark F is the point under the reflector R that is illuminated by the lamp L_a. The fiduciary mark is the diametric line of a circle having two ears (Fig. 94), the whole constituting the image. The figure shows the three useful images and the scale of the reticule.

The three images are—

1. *The Reading Image,* or "steady" image, which is obtained when the auto-collimator beam is reflected from the roof surface of the prism A (Fig. 92). It lies on the scale and its

position relative to the scale is governed only by the rocking of the lozenge and prism, and is not affected by relative motion between the specimen and the auto-collimator in any direction

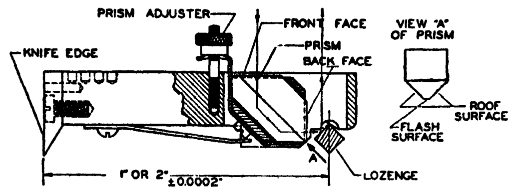


FIG. 92. DETAIL OF GAUGE ELEMENT

except a rotation about the longitudinal axis of the auto-collimator. In the latter case the image is merely shifted to a position above or below the scale and parallel to it. With the fiduciary mark is shown the vernier scale for reading to 0·1 of a graduation.

2. *The Flash Image*, shown immediately above the "steady" image, is obtained when the auto-collimator beam is reflected from the flash surface of the prism (Fig. 92). This image travels parallel to the steady image but it may be displaced vertically above or below the scale by rotating the extensometer. The flash image is used chiefly to eliminate errors due to rotation of the extensometers; hence it must appear in the field of view whenever a reading of the "steady" image is made.

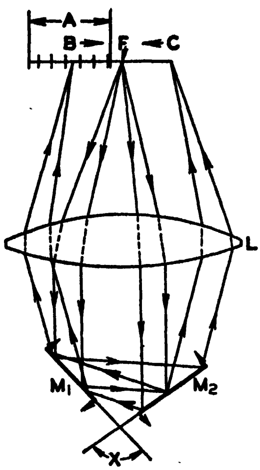


FIG. 93. ILLUSTRATING THE OPTICAL PRINCIPLE OF THE TUCKERMAN STRAIN GAUGE

3. *The Front-face Image*, shown to the right of the flash image, which is obtained when the auto-collimator beam is reflected back by the front face of the prism. It normally serves no useful purpose and is usually positioned by rocking the auto-collimator so that it does not interfere with the reading of the steady image. In certain problems, however, such as those involving vibration, this image can be used to obtain the qualitative measurement of the direction and nature of the vibration.

This image is usually displaced from the scale by the same amount as the flash image, but unlike the latter may be moved

parallel to the scale only by rocking the prism. It may be shifted with respect to the scale by rotating the extensometer about its longitudinal axis, in which case it will travel parallel to the flash image.

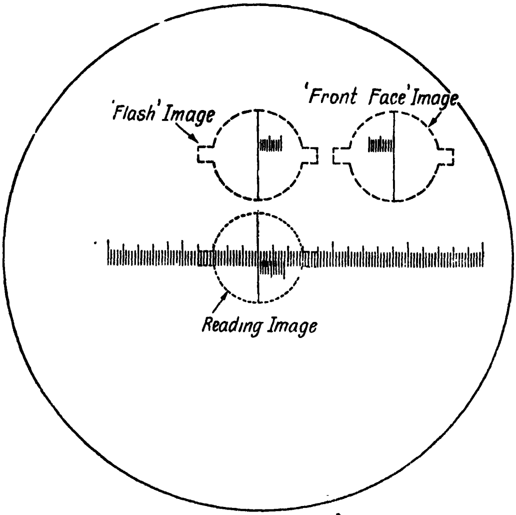


FIG. 94. SHOWING THE SEVERAL IMAGES SEEN THROUGH THE EYEPIECE OF THE AUTO-COLLIMATOR

The auto-collimator need not be mounted in a fixed relationship with the extensometer or at any fixed distance from it.

Factors for converting the gauge readings to strain values are supplied with the instruments.

Autographic Recorders. The Riehle recording gear is shown in Fig. 95. The machine itself is a multi-lever machine similar to the Olsen machine which with its autographic gear was described in Chapter III.

The diagram shows the autographic and automatic gear, the specimen, and part of the poise screw, all rotated into the same plane in order to make the mode of operation clear.

A micrometer dial *M* is fixed to the poise screw to indicate small increments of load. A bevel pinion on the spindle operating the poise screw transmits motion to the vertical screw *V* and operates the nut which carries the pencil. The movement of the pencil along the drum is proportional to the movement of the poise weight and provides the load ordinate.

The drum is rotated in the following manner.

The bracket carrying the rod *R* is bolted to one of the columns on the platen of the machine. A telescopic slide moves on the rod *R*. The slide consists of three parts: an inner tube *A*, which slides on the rod; an intermediate tube *B* which slides on the first; and an outer tube *C*, provided with friction rollers, free to slide on the tube *B*. Each can be clamped in position by means of a thumb screw.

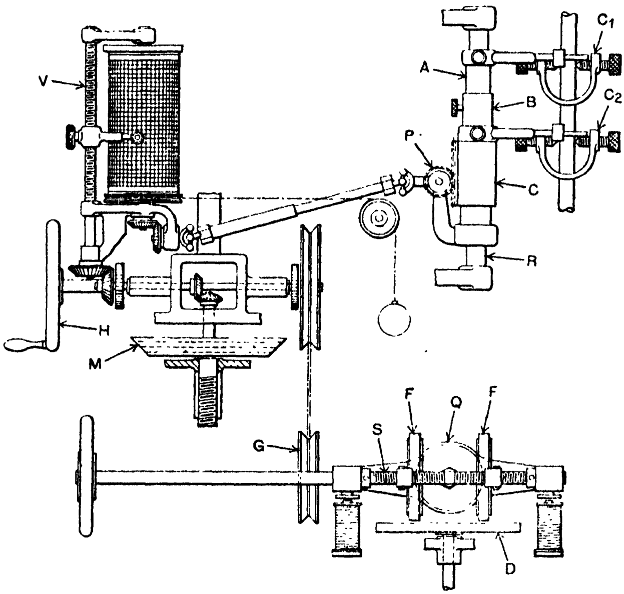


FIG. 95. RIEHLE AUTOGRAPHIC RECORDING GEAR
(*Machinery*)

The tubes *A* and *C* each carry an adjustable fork which rests on a clip pinched on to the specimen at a gauge point. Under operating conditions the inner tube *A* is allowed to slide on the vertical rod by virtue of the weight of the parts and is supported by the fork resting on the upper clip C_1. The intermediate tube is clamped to *A* by its screw, while the outer part *C* is left free on *B* and supported by the fork resting on the lower clip C_2 on the test bar.

Any movement between the top grip and the upper gauge point results in the tubes moving bodily down the rod with no

relative movement. Extension of the specimen results in the tube C moving relatively to A by an amount equal to the extension between the gauge points. The tube C carries a rack which drives the pinion P and this rotates the chart drum through a double Hooke's joint. The rotation of the drum provides the strain ordinate. The magnification is 5: 1.

Automatic operation is obtained by means of a friction drive. The disc D is driven at a suitable speed by a light leather belt not shown. Two fibre discs F, F are secured by a feather to a spindle working in bearings in the swivel plate Q which is supported by the frame of the machine. The swivel is free to oscillate about the vertical axis so as to permit one or other of the fibre wheels to engage the driving disc. When either engages, the spindle is rotated and operates the poise screw in the appropriate direction. This is accomplished by the belt drive from the pulley G. The swivel carries two armatures or keepers, and is controlled by electromagnets, one of which is put in circuit whenever the beam makes contact with the top or bottom stop. The rate of movement of the poise can be regulated by hand by means of the screwed spindle S. This carries two forks which bear on the fibre discs and can be moved radially across the face of the driving disc with consequent alteration of the velocity ratio. When everything has been set up, the clutch operating the straining gear is thrown in and a switch is closed to complete the circuit when the beam touches a contact point. After closing the switch the test proceeds automatically.

The Denison Autographic Recorder, Figs. 96 and 97, is made in two types. The drum carrying the chart is of aluminium and is mounted on a ball bearing spindle. The largest size of graph is 10 in. × 10 in. A stepped pulley having flat grooves for cord driving is mounted on the spindle and provides the choice of several scales. The pulley drives the drum by friction, a method that facilitates the setting of the pen to zero. The motion of the drum is obtained from the extension of the specimen, clips being attached at the gauge points to take the driving cords. The stress ordinate is provided by the travel of the pen parallel to the drum axis. The pen is carried on a duralumin bar which rests on two broad-faced pulleys mounted on ball bearings. A taut wire attached to the bar passes round the pulleys and ensures that all move in unison. The arrangement is clearly seen in Fig. 97.

The stress ordinate in one type of recorder is obtained from the travel of the poise along the beam. The test is conducted in the ordinary way, the poise weight being traversed to maintain equilibrium.

For single-lever machines a compensating mechanism is fitted which compensates for fluctuations of the steelyard within

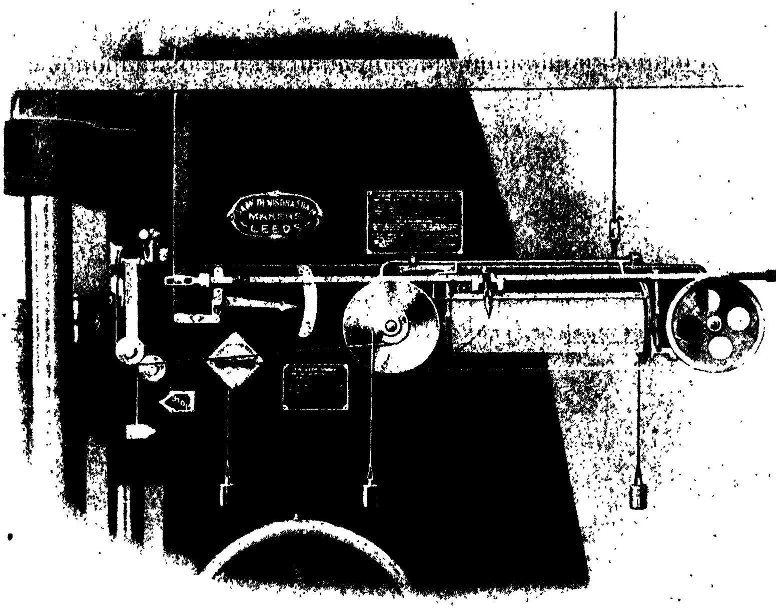


FIG. 96. DENISON AUTOGRAPHIC RECORDER

The steelyard balance indicator is visible to the left and the rod of the compensating gear can be seen behind the right hand end of the drum.

(Denison & Son Ltd.)

about $\pm$ 3 per cent of the full capacity of the machine. This is an important feature since it is somewhat difficult to keep the steelyard floating exactly in its mid position. The limits of automatic compensation and the exact position of the steelyard are shown by the indicator.

This type of recorder is the more accurate as it works off the lever and deadweight system of the machine.

In the other type the stress ordinate is produced from the

extension of one or more calibrated springs which serve as a load resistant and which automatically maintain equilibrium. Any number of scales may be obtained by the use of appropriate springs. With this type the recorder itself gives the only indication of the test. When fitted to multiple lever machines

FIG. 97. FRONT VIEW OF DENISON SINGLE-LEVER VERTICAL TESTING MACHINE

With cords in position over specimen frames ready to commence test.

(*Denison & Son Ltd.*)

it is provided with a dashpot for damping out the oscillations of the spring. Means are provided for calibrating the springs.

Other recorders employing a calibrated spring are the Wicksteed and the Buckton-Wicksteed in which the principle is the same as in the instrument just described.

Amsler Recording Extensometer. Amsler's recording extensometer used in conjunction with the Pendulum Dynamometer (page 55) is shown in Fig. 98.

The instrument consists of two knife-edged grips which are held together by two telescopic tubes fitting into each other. Each knife-edged grip is provided with a sharpened disc, with two notches, to bear on the test bar at the gauge point. The round part of the disc is used for attaching the instrument to flat specimens while the notches are used for gripping round bars.

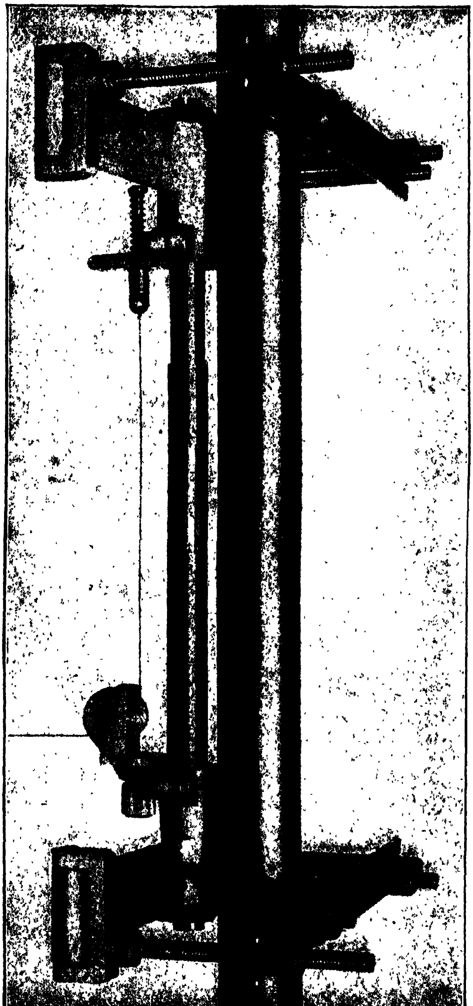


Fig. 98. Amsler Recording Extensometer
(*Alfred J. Amsler & Co.*)

A set of telescopic tubes allows gauge lengths of 8 in., 6 in. and 4 in. to be employed.

When the specimen extends the elongation is transmitted to the recording drum by an inextensible cord. The end of the cord is fixed to an eyelet mounted on one of the grips of the extensometer, the eyelet being kept in position by a spring. The cord next passes round a guide pulley mounted on the other grip and from there is led to the recording drum for the diagram.

The eyelet to

which the cord is attached prior to the test gives the cord an initial tension which is effectively maintained as the test bar is stretched.

The grip carrying the guide pulley should always be attached to the test bar end nearest that gripping head of the machine which does not move during the test.

The tubes forming the connection between the two grips are provided with a scale and an indicator showing the reading at every moment of the deformation undergone by the bar. The extensometer is unaffected by the shock when the test piece breaks.

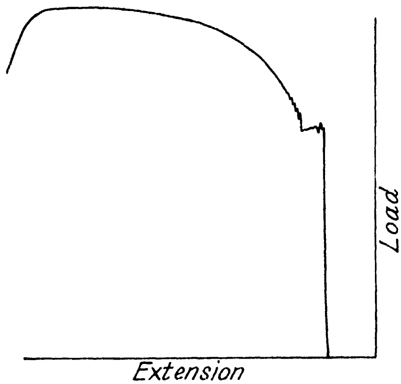


FIG. 99. DIAGRAM TAKEN ON AMSLER MACHINE
(Alfred J. Amsler & Co.)

When using plain paper for the diagram, as advocated by Messrs. Amsler, it is advisable, just before starting a test, to mark the zero and maximum lines by holding the pointer on the dial, first at zero and then at the maximum of the load range, and in each position revolving the recording drum with the free hand. The distance between these two basic lines (4 in.) then represents the full capacity of the corresponding range and any intermediate position of the diagram can be found by using a draughtsman's scale.

A diagram taken on an Amsler machine is reproduced in Fig. 99.

Messrs. Amsler have developed a new type of recorder, magnifying fifty times and thus giving an enlarged scale for the elastic portion of the stress strain diagram. This is accomplished by means of a micrometer screw with a large head supported by the outer guide tube. The screw is pointed and touches a stop attached to the inner guide tube. One end of a thread wound round the head of the micrometer is led over a fixed guide roller and is loaded with a small weight—the driving weight. The other end of the thread runs over guide rollers on the extensometer and over the pulley on the recording drum. This also is loaded with a weight—the tension weight.

During the elongation of the test bar the upward movement of the upper clamp sets the micrometer screw out of contact with the stop attached to the lower clamp. The micrometer screw, which is thereby set free, revolves under the action of the driving weight, which is heavier than the tension weight, and screws itself down until the point again touches the stop. These movements are transmitted to the recording drum by means of a thread.

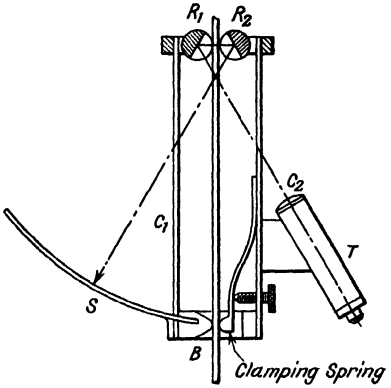


FIG. 100. WESTINGHOUSE EXTENSOMETER FOR WIRES

By leading the thread over the large pulley on the recording drum the magnification can be increased to 100 times. The graph obtained in this way permits the modulus of elasticity, the limit of proportionality, and the yield point to be ascertained.

As such a large magnification would be useless beyond the yield point an arrangement to limit the play is provided. This automatically changes the magnifying extension into an ordinary recording extensometer, the elongations being recorded full size if the thread is led over the large pulley, or double size if it is led over the small pulley on the arm.

This automatic change-over takes place as soon as the elongation of the test bar has reached a dimension corresponding to the yield point which may be expected.

A suspension arrangement prevents the instrument from falling when the test piece fractures. The instrument, which must be used in a vertical position, can be adapted to round bars from $\frac{1}{16}$ in. to 1 in. diameter.

The Westinghouse Extensometer. The Westinghouse extensometer for wires is shown in Fig. 100. It consists of two clips C_1 and C_2 fastened at one end to a block B provided with a slot for the test wire to pass through. The free ends of the clips carry hardened steel rollers R_1, R_2, mounted on pivots. The rollers carry plane mirrors and a spot of light from a lamp in the tube T is projected on to one of the mirrors and

passes by reflection to the scale *S*. Alterations in the length of the test wire of 0·00002 in. can be measured.

Dalby's Optical Recorder. Designed to obviate inertia errors, this instrument is shown in Fig. 101. The load measurement is obtained from the extension of a steel bar, the bar

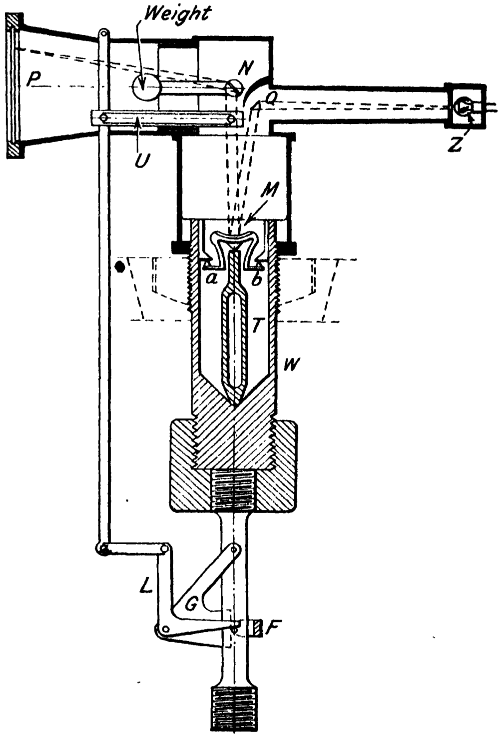


FIG. 101. DALBY'S OPTICAL RECORDER

acting as a stiff spring. The bar *W* is hollow and is gripped at its upper end in the head of the machine. A small concave mirror *M* is free to move about the axis *ab*, the tilt being effected through the medium of the steel tube *T*.

The beam of light from the lamp *Z* is reflected by the mirror *Q* on to the mirror *M* and by a third mirror *N* to the photographic plate at *P* where the beam is focused to move horizontally across the plate. The magnification is of the order of 340.

One axis of the mirror *N* is connected to the specimen *S* by the linkage shown. This consists of a frame *F* carrying two points which are clamped to the lower gauge point and a frame *G* which carries the clamps for the upper gauge point.

By means of the bell-crank lever *L* and the associated linkwork which is connected to the axis of the mirror *N* through an arm and the rod *U*, the mirror is tilted when extension of the test piece occurs. The spot of light is thus caused to move vertically over the photographic plate proportionately to the extension of the specimen.

Records of exceptionally rapid tests can be made with this apparatus and it is possible to obtain complete load-extension diagrams in 10 sec.

CHAPTER VII

TORSION TESTING MACHINES

TORSION tests are carried out to determine the modulus of rigidity of a material—the ratio of shear stress to shear strain—or to ascertain its ultimate torsional strength.

Tests are made, most commonly, on specimens of round section. This is the only section to which the elementary theory of torsion is applicable.

The torque T which will induce a stress f in a bar of diameter d is

$$T = (\pi/16)d^3f \quad . \quad . \quad . \quad \text{(see page 5)}$$

When a solid bar of mild steel is tested in torsion it is found that the yield point is less marked than in the tension test. Failure is masked owing to the fact that the whole of the material is not at once stressed to the same degree. The stress is greatest in the outer layers and becomes more and more uniformly distributed over the section as the plastic stage is traversed. There is no appreciable reduction in the cross-section of the test piece and hence the stress-strain curve does not show the pronounced droop at the end as in the tension test, but becomes almost parallel to the strain axis. The shear stress being then uniformly distributed, the maximum stress is calculated by the formula

$$T = (\pi/12)d^3f$$

where T is the breaking torque.

In making a test, it will be found that after the true elastic stage has been passed, the steelyard, for each increment of load, tends to fall due to creep, and that only approximately accurate readings can be taken. Equal intervals of time should be allowed between each successive pair of readings.

In order to determine the yield stress more accurately the test piece is sometimes made in the form of a tube. The stress distribution across the comparatively thin wall of the tube being more nearly uniform, the yielding will be more definite. Thin-walled tubes are not, however, very suitable for the determination of the maximum strength as they frequently deform after creep has begun.

The modulus of rigidity G can be found from the formula

$$G = 32Tl/\pi d^4\theta \qquad . \qquad . \qquad . \text{ (see page 6)}$$

when the gauge length l, the diameter d of the specimen, and the angle of twist θ produced by the applied torque T are known.

In the case of tubes of outside diameter D and inside diameter d, the foregoing expressions become respectively

$$T = \frac{\pi}{16}\frac{(D^4 - d^4)}{D}f$$

and

$$G = \frac{32\,Tl}{\pi(D^4 - d^4)\theta}.$$

Ductile materials, such as mild steel, fracture on a plane at right angles to the axis of the test piece, but brittle materials fracture along a 45° helix, indicating that failure occurs where the tensile stress is a maximum.

The usual method of applying a torque to the specimen is by means of a worm and wheel driving a chuck in which one end of the test piece is gripped. The other end of the test piece is held, either in the steelyard directly, or in one arm of a multiple-lever system. The applied torque is balanced by the moment of the poise about the fulcrum of the weighing lever.

While many universal testing machines have provision for making torsion tests it is often advantageous to employ a separate machine for this purpose. A good example is the machine made by Messrs. W. & T. Avery Ltd., which permits tests to be made in reverse torsion.

The machine is illustrated in Fig. 102. The specimen is gripped in chucks *A* and *B*. Torque is applied by turning the handwheel *H* which drives through the worm gear *F*. The floating sleeve carrying the chuck *A* is free to slide longitudinally and so prevents an end load from being imposed on the test piece as twisting proceeds.

The torque is transmitted by the chuck *B* to the cross-lever *L* fitted with knife-edges *E* at its ends.

According to the direction of the twist, the main lever pulls on one of the intermediate levers which in turn communicates the pull to the steelyard. The latter carries the balance weight *D* and the movable poise *w* operated by the small handwheel *J*. The main lever has a fulcrum consisting of hardened steel cones and cups bearing upon rings of hard steel balls. The

connection between the handwheel, which is on a fixed portion of the frame, and the gearing on the steel yard is made at the point of no motion, that is, at the fulcrum; consequently, the pressure upon the handwheel is not communicated to the steelyard. This obviates the possibility of the specimen being broken prematurely. The steelyard is fitted with a polished scale plate graduated from zero to 2 000 lb.-in. by 50 lb.-in. divisions:

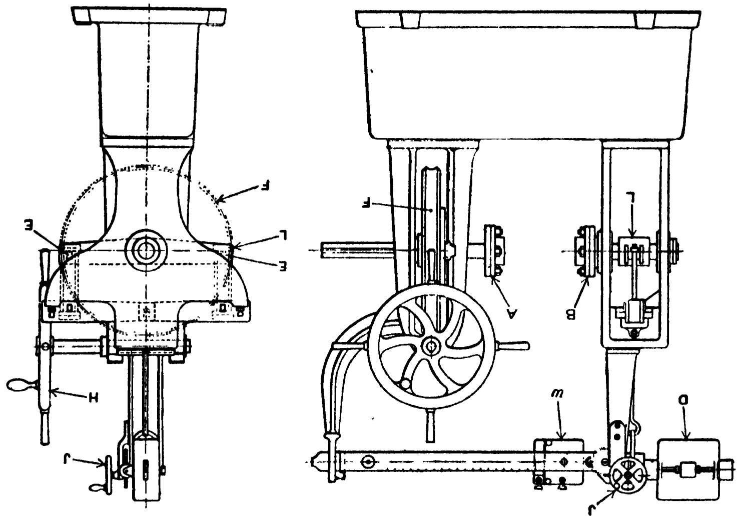


FIG. 102. AVERY TORSION TESTING MACHINE
(*Machinery*)

a vernier scale on the poise subdivides these into divisions of 5 lb.-in.

The full capacity of the machine is 10 000 lb.-in. and additional poise weights are supplied for this capacity. When the poises are combined the scale readings need to be multiplied by 5.

A graduated ring mounted on the wormwheel enables angles of twist on the whole length of specimen to be measured over the plastic stage. The specimen used in this machine has coned ends each provided with two keyways, as shown in Fig. 103. The maximum length of specimen between shoulders is about 10 in.

As this type of specimen is somewhat expensive to make, it is an advantage, for ordinary work, to have additional holders to take specimens having square ends.

For measurements within the elastic stage some form of torsion strain meter is needed. Several commercial instruments are available for making such measurements.

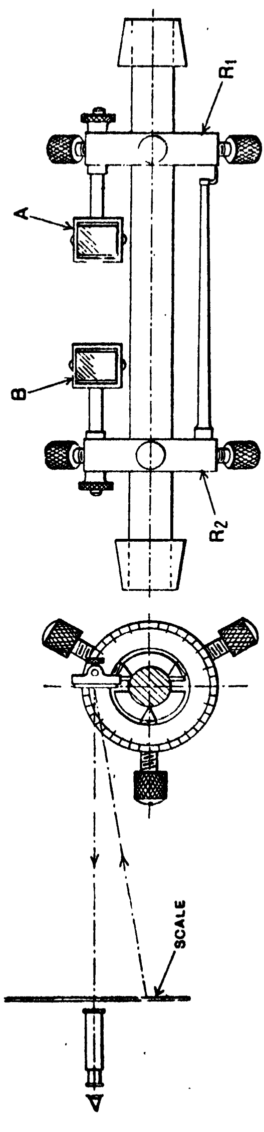


Fig. 103. Simple Form of Mirror Torsion Meter (*Machinery*)

A simple and effective arrangement by which the strains can be observed is shown in Fig. 103. Two rings, R_1 and R_2, each carrying a mirror (A, B) which can be tilted about two axes at right angles, are clamped to the gauge points. These, used in conjunction with two scales and two telescopes, enable observations of the angular displacement to be made. If the two mirrors are arranged close together it is possible to work with only one telescope and one scale.

Measurements of the twist during the plastic stage may be made if one ring is provided with a scale of degrees and the other with a suitable pointer the end of which traverses the scale on the first ring.

The table on page 139 gives the results of a test on a mild steel specimen $\frac{7}{8}$ in. diameter and 6 in. between gauge points.

Referring to Fig. 103, suppose the mirror A to be adjacent to the end of the test piece to which the torque is applied, while the other, B, is near the end gripped by the main lever. In the table, column 3 contains the observations taken on mirror A and column 5 those taken on mirror B. If the initial reading, namely 16, of column 3 be subtracted from the remaining numbers in the column

TABLE VII

RECORD OF TEST OF A MILD STEEL BAR ⅞ IN. DIAMETER

No. of Reading (1)	Applied Torque T lb. in. (2)	Observed Reading Mirror A (3)	Corrected for Zero (4)	Observed Reading Mirror B (5)	Corrected for Zero (6)	Twist on 6-in. Gauge Length (7)
1	—	16·0	—	10·3	—	—
2	850	26·6	10·6	14·7	4·4	6·2
3	1 100	28·6	12·6	15·5	5·2	7·4
4	1 225	29·9	13·9	15·7	5·4	8·5
5	1 350	31·25	15·25	15·8	5·5	9·75
6	1 425	31·5	15·5	15·9	5·6	9·9
7	1 475	31·9	15·9	15·8	5·5	10·4
8	1 600	33·1	17·1	16·2	5·9	11·2
9	1 720	34·2	18·2	16·6	6·3	11·9
10	1 820	35·4	19·4	16·8	6·5	12·9
11	1 975	36·5	20·5	17·25	6·95	13·55
12	2 100	37·7	21·7	17·25	6·95	14·75
13	2 350	39·8	23·8	17·7	7·4	16·4
14	2 600	41·7	25·7	17·65	7·35	18·35
15	2 850	44·3	28·3	18·5	8·2	20·1
16	3 100	46·2	30·2	18·6	8·3	21·9
17	3 150	56·7	40·7	19·7	9·4	31·3

and tabulated in column 4, the values so obtained represent the twist up to the ring R_2. The differences between corresponding numbers in columns 4 and 6 give the values of the twist, in scale divisions, on the given gauge length. These values are tabulated in column 7.

The distance between the mirror and the scale being denoted by L, the angle of twist θ radians is found from the formula

$$\tan \theta = \frac{\text{twist in scale divisions}}{2\,\mathrm{L}}.$$

In most cases it is sufficiently accurate to replace $\tan \theta$ by θ itself. For an angle of 5° the difference between the radian measure of the angle and its tangent is, from four-figure tables, 0·0002 and for an angle of 10° the difference is 0·0018.

In the present instance the scale readings are in centimetres and the distance between the scale and mirror is 12 ft. 6 in. Hence the multiplying factor for converting the readings in column 7 into radian measure is

$$\frac{1}{2 \times 12{\cdot}5 \times 12 \times 2{\cdot}54} = 0{\cdot}001313.$$

If, in the setting up of the test piece, a small initial torque be inadvertently applied, the correct value of the torque can be ascertained at the end of the test by plotting the torque-twist diagram and producing the straight portion of the graph to intersect the vertical axis through the origin. The magnitude

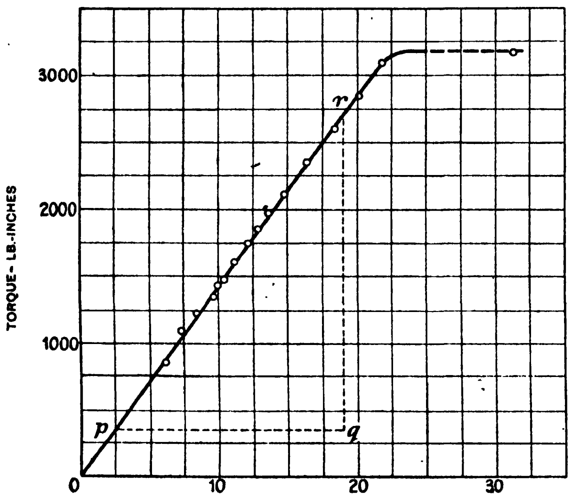


FIG. 104. TORQUE-TWIST DIAGRAM FROM TEST OF $\frac{7}{8}$ IN. DIAMETER MILD STEEL BAR

(Machinery)

of the intercept will give the value of the initial torque. The diagram can then be replotted, if desired, with the correct zero.

The results in columns 2 and 7 are plotted in Fig. 104.

To determine the modulus of rigidity we have

$$G = 32Tl/\pi d^4\theta$$
$$= (32l/\pi d^4) \times (T/\theta)$$

The ratio T/θ is given by the slope of the graph. Thus

$$G = \frac{32l}{\pi d^4} \cdot \frac{T}{\theta}$$
$$= \frac{32 \times 6}{3{\cdot}1416 \times (0{\cdot}875)^4} \cdot \frac{\overline{qr}}{0{\cdot}001313\overline{pq}}$$

$$= \frac{32 \times 6}{3{\cdot}1416 \times (0{\cdot}875)^4} \cdot \frac{2\ 350 \text{ lb.-in.}}{0{\cdot}001313 \times 16{\cdot}5 \text{ scale divisions}}$$
$$= 11\ 300\ 000 \text{ lb. per in.}^2$$

Fig. 105 shows the results of a torsion test on a mild steel bar pierced transversely.

In the torsion machine (Fig. 106), made by Messrs. Alfred J. Amsler & Co., of Schaffhouse, the load is measured through the

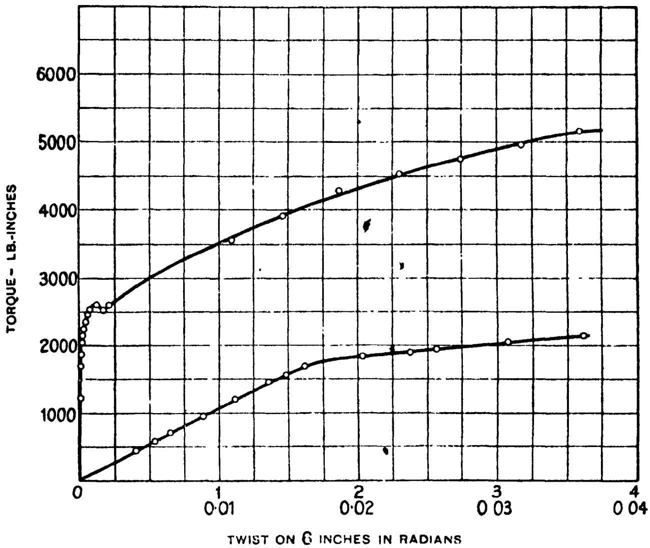


FIG. 105. TORQUE-TWIST DIAGRAM OF TEST OF $\frac{7}{8}$ IN. DIAMETER MILD STEEL BAR PIERCED TRANSVERSELY
(*Machinery*)

medium of a pendulum. The gripping head at the left of the machine is fixed to the end of a shaft which rotates in ball bearings and is rigidly connected to the pendulum rod. When the test bar is in position the torque applied to the worm drive tends to rotate the pendulum, which is then deviated from its position of equilibrium to such an extent that its static moment balances the applied torque.

To prevent the pendulum from falling back too quickly at the instant when the test bar fractures, it is hooked on to a brake rope which prevents a too rapid descent. The inclination of the pendulum is transmitted to the spindle of a pointer which indicates on a dial the moment of torsion in lb.-ft.

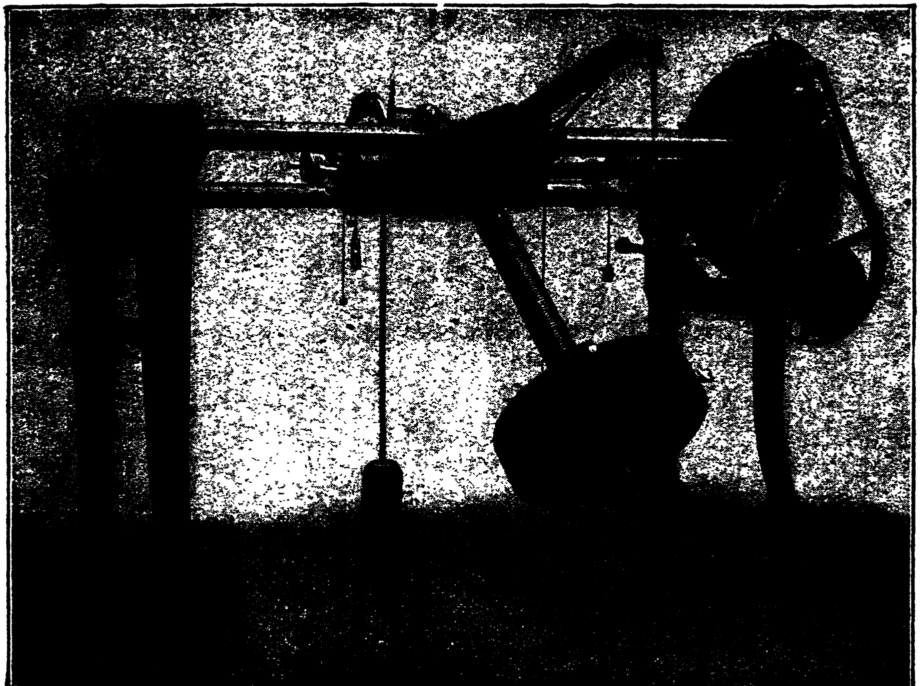


FIG. 106. AMSLER PENDULUM TYPE TORSION TESTING MACHINE
(*Alfred J. Amsler & Co.*)

A record of the test is given by an autographic device similar in principle to that described in Chapter VI.

The deflection couple of the pendulum can be changed by altering the position of the bob and a separate dial is provided for each degree of sensitivity produced.

For gripping the test bar, taper-wedge grips are provided, smooth wedges being used for flat specimens and toothed wedges for round bars. In making torsion tests on smooth

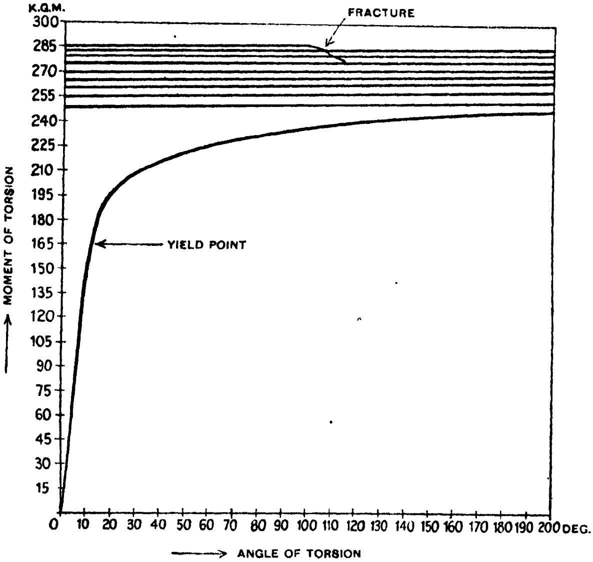


FIG. 107. AUTOGRAPHIC RECORD TAKEN ON AMSLER MACHINE (*Machinery*)

round bars the ends are enclosed with wood slats which are gripped in the toothed wedges. When testing tubes, it is necessary to fit plugs in the ends in order to prevent the tubes from being flattened at the gripping point. Special gripping heads are provided for dealing with crankshafts when it is required to test these in torsion.

The twist on a given gauge length is measured by means of two graduated rings and two discs moving inside the rings. The rings are clamped to the horizontal bars of the machine and the discs to the specimen at the gauge points. The angle of torsion can be read to $\frac{1}{5}°$.

An autographic record taken from one of these machines is reproduced in Fig. 107. The specimen tested was a round bar 29 mm. diameter with a gripping length of 430 mm. It twisted 5·28 revolutions before fracture.

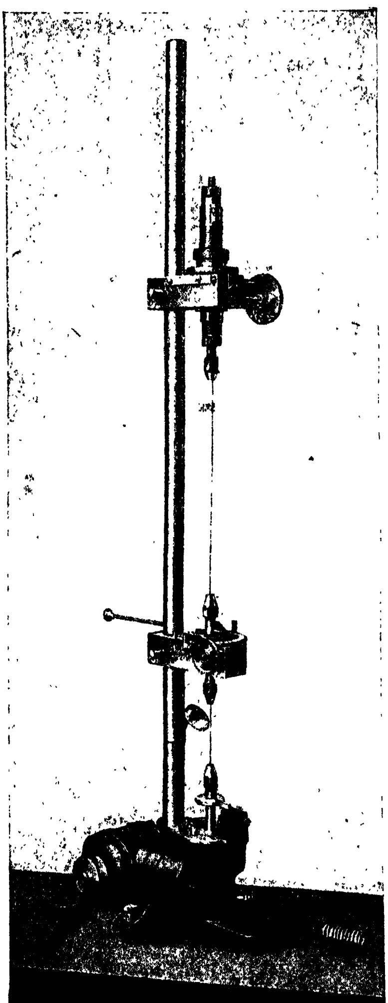


FIG. 108. AMSLER MACHINE FOR TESTING WIRES IN TORSION
(Alfred J. Amsler & Co.)

A machine for testing wires in torsion is shown in Fig. 108. The apparatus consists of a column carrying three arms. The wire to be tested is secured between the middle and lower arm, the ends of the wire being held between wedges in the clamping heads. A calibrated wire—the measuring wire—which twists with the wire under test, serves for measuring the moment of torsion. It is placed between the middle and upper arms and is secured to the upper arm and to the spindle of the upper clamping head.

The longitudinal tension on the wire can be regulated by adjusting a tension spring: two such springs are supplied, one for tensions of from 0 to 10 lb. and one for tensions of from 0 to 40 lb.

The spindle of the lower clamping head for the specimen wire carries a toothed wheel which is set in rotation by turning a handle. The number of twists applied to the test wire up to its breaking point is

indicated on a counter actuated from the drive of the lower clamping head. Angles of twist may be measured to one-hundredth of a revolution.

The angle of torsion of the measuring wire is shown on a circular scale mounted on the intermediate arm. A driving pin which is inserted in the spindle and to which is secured the lower clamping head of the wire under test, pushes a loose pointer over the circular scale. The distance traversed by this pointer depends on the torsional resistance of the measuring wire, and in consequence is a measure of the torsional moment applied to the test wire.

The angle of torsion of the measuring wire as a function of the angle of torsion is determined by experiment.

The device for calibrating the measuring wire consists of weights which exert equal and opposite forces on the ends of a cross-pin fixed to the spindle of the lower head of the measuring wire. For calibrating very thin measuring wires the checking weights are suspended according to the bifilar principle. The calibrating arrangement, which is only needed occasionally, is removed from the apparatus during torsion tests.

Four measuring wires are provided of 50, 5, 0·5 and 0·05 lb.-ft. capacity.

The free length of the test specimens between clamping heads is 4 in. or 2 in., according to the settings of the upper and middle arms. Wires from the thinnest up to $\frac{9}{64}$ in. diameter can be gripped. The illustration shows the machine arranged for motor driving. In this case the lower clamping head rotates at about 20 r.p.m.

CHAPTER VIII

HARDNESS TESTING

Hardness. An important property of a material is its hardness, a property easily comprehended in a general way; but one that eludes precise definition and whose existence as a definable quality is doubted.

The usual interpretation placed upon the term hardness is that of resistance to penetration, or resistance to abrasion. But these views are quite distinct since a material may offer considerable resistance to abrasion and yet be relatively soft according to standards based on indentation tests.

Hardness tests, as usually made, involve the breakdown of the material tested. The result is that the distribution of stress is not known with certainty and hardness values are therefore based on the consequences of the stresses produced.

Although such tests are merely relative they are of great value to engineers. Primarily, hardness, however it may be understood, is important in that it enables a material to withstand certain conditions of service. Its importance in the scheme of mechanical testing lies in the fact that its quantitative determination affords an estimate of the tensile strength of steel and wrought iron and, moreover, throws light on the treatment which the material has received during manufacture. The ease with which hardness tests can be made has led to their widespread adoption in commercial testing.

Scratch Tests. The mineralogical scale of hardness is based on a scratch test and consists of a number of substances arranged in an empirical series. The arrangement indicates that each substance will scratch the one preceding it in the scale but not the one that succeeds it. The scale, usually termed *Moh's Scale*, is as follows—

1. Talc.	6. Felspar.
2. Gypsum.	7. Quartz.
3. Calcspar.	8. Topaz.
4. Fluorspar.	9. Corundum.
5. Apatite.	10. Diamond.

Here there is no attempt at measurement, the relationship being purely qualitative.

The scratch test has, however, been developed by Martens and other workers and tests have been introduced in which a diamond, loaded by a movable poise on a lever, scratches the test piece. Hardness is then defined as the load in grams under which a conical diamond would produce a scratch 0·01 mm. wide. In recent years further attention has been given to "scratch hardness" but this aspect of hardness testing will not here be pursued further. Instead, attention will be devoted to the indentation test, as this is now standardized and plays an important part in the workshop testing of metals.

Indentation Tests. Indentation tests are made under static conditions and consist in forcing a ball or pointed body into the test piece under a dead load. Conical and pyramidal points, as well as spheres, are used. The former offer some advantage, but the general run of hardness tests are, at present, made with a ball.

In the *Brinell method*, from which the test takes its name, a steel ball is forced into the test piece under pressure and the hardness of the material is expressed as a number—the Brinell hardness number—which is defined as the stress per unit of spherical area.

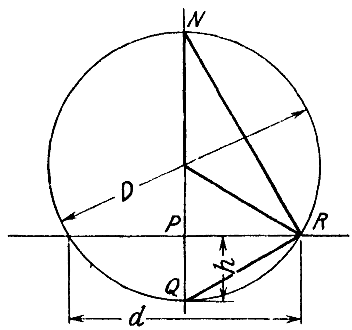


FIG. 109. ILLUSTRATING METHOD OF DERIVING THE BRINELL HARDNESS NUMBER

If P is the load applied in kilogrammes and A the spherical area in square millimetres, the hardness number H, is given by

$$H = P/A = P/\pi Dh$$

where D is the diameter of the indenting ball and h the depth of the indentation produced, measured as indicated in Fig. 109.

If d is the diameter of the impression we have from the similar triangles NPR, RPQ, in the figure,

$$PQ/PR = PR/NP$$

or

$$\frac{h}{d/2} = \frac{d/2}{D - h}$$

from which $h^2 - hD + d^2/4 = 0$

Consequently $$h = \frac{D \pm \sqrt{(D^2 - d^2)}}{2}$$

and for the depth of the impression to be less than half the diameter of the ball the minus sign must be taken, giving

$$h = \tfrac{1}{2}[D - \sqrt{(D^2 - d^2)}].$$

By a rule of mensuration the area A of the spherical surface is πDh, that is

$$A = \pi(D/2)[D - \sqrt{(D^2 - d^2)}]$$

and the hardness number calculated on the diameter of impression as a basis, is

$$H = \frac{P}{(\pi D/2)[D - \sqrt{(D^2 - d^2)}]}.$$

In cases where it is necessary to make the numerical calculation the expression may be put into a more convenient form thus—

$$(D/2)[D - \sqrt{(D^2 - d^2)}] = (D^2/2)[1 - (1 - d^2/D^2)^{\frac{1}{2}}]$$

$$= \frac{D^2}{2}\left[1 - \left(1 - \frac{1}{2}\frac{d^2}{D^2} - \frac{1}{2}\cdot\frac{1}{4}\left(\frac{d^2}{D^2}\right)^2 - \frac{1.1.3}{2.4.6}\left(\frac{d^2}{D^2}\right)^3 \cdots\right)\right]$$

$$= \frac{d^2}{4}\left(1 + \frac{d^2}{4D^2}\right)$$

neglecting the sixth and higher powers of d/D.

So that

$$H = \frac{P}{\frac{\pi d^2}{4}\left(1 + \frac{1}{4}\cdot\frac{d^2}{D^2}\right)}.$$

The error involved by the use of this approximation is less than 2 per cent even when the ratio $d/D = 0{\cdot}6$, the largest value advised in ordinary testing.

It is customary, whenever possible, to employ a load of 3 000 kg. and a ball 10 mm. diameter, The diameter of the impression is measured by means of a microscope. In some forms of instrument the microscope carries a cross wire and is moved bodily over the field of view by a micrometer screw. The cross wire is first brought into coincidence with one edge of the impression and the reading of the micrometer dial noted. The cross wire is then brought into coincidence with the opposite

edge of the impression and the reading again noted. The difference between the two readings gives the diameter of the indentation.

In other instruments the image of the impression is focused on a scale in the eyepiece whereby the diameter can be read off directly.

Two measurements of the indentation should be taken at right angles to each other and the mean of the two values used in calculating the hardness number.

For commercial testing the microscope should be capable of measuring the diameter of the impression to 0·05 mm.

The surface of the test piece should be smoothly finished and if a small ball of 1 or 2 mm. diameter is being used the surface of the specimen should be brought to a polish. No. 0 emery cloth is satisfactory for loads of 30 kg. or more. For very small loads the finish should be with No. 00 or 000.

The British Standard Specification requires that the centre of the impression shall be not less than two and a half times the diameter of the impression from any edge of the test piece, and that the thickness of the test piece shall be such that no marking showing the effect of the load shall appear on the underside.

The value of the hardness number is affected by the diameter of the ball, by the pressure, and by the distortion of the ball itself. On the latter account it is recommended that with hardness numbers above 500 care should be taken to see that the balls are considerably harder than the material tested.

Experiments by Meyer showed that the mean pressure per unit area was constant when the indentations were geometrically similar; that is, $4\,P/\pi d^2 =$ a constant.

It follows that as d/D and $4P/\pi d^2$ are constant, for similar indentations on the same material, P/D^2 is also constant. The importance of this relation is that it enables comparative tests to be made where a 10 mm. ball and a load of 3 000 kg. are not applicable.

Thus, if with a small test piece, a ball of diameter D_1 is used, the corresponding load P_1 that should be applied is—

$$\text{For a 5 mm. ball } P_1 = 30 \times 5^2 = 750 \text{ kg.}$$

$$\text{For a 1 mm. ball } P_1 = 30 \times 1^2 = 30 \text{ kg.}$$

The operation of indenting the material results in a ridge

being formed around the impression, Fig. 110 (*a*), or in a depression as in Fig. 110 (*b*). The former is noticeable in copper and mild steel; the latter in manganese steel and some bronzes.

As the hardness numbers, as calculated from the curved area of impression, are not strictly comparable for different materials, the depth of the indentation below the original surface has been suggested as forming a more rational basis of comparison. This, however, is not the basis of the Brinell hardness number according to the British Standard Specification, which stipulates that the hardness number must be calculated from the diameter and not from the depth of impression.

The hardness numbers calculated from the diameter of the impression are not independent of the load, but if the depth

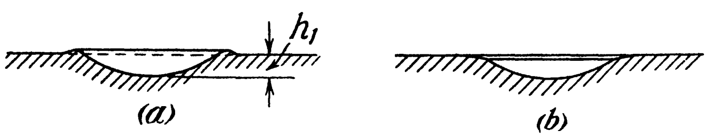


FIG. 110. TEST INDENTATIONS
(*a*) Showing ridge round the edge of the indentation.
(*b*) Showing depression round the edge of the indentation.

from the original surface h_1 be used, the numbers obtained are, for some materials, independent of the load. Carrington has shown, however, that there are considerable divergences from this rule.

For steel and materials of a like degree of hardness, a load of 3 000 kg. with a ball 10 mm. diameter is employed. This makes the ratio $P/D^2 = 30$ as shown previously. For copper alloys the specified value of P/D^2 is 10, while for copper it is 5 and for lead and tin it is unity.

The effect of time on the size of the impression formed has little effect after the first 10 sec., at least for steels. The minimum time of application of the load given by the specification is 15 sec. when $P/D^2 = 30$, and 30 sec. when $P/D^2 = 10$, 5 or 1. The rate of application of the load may, however, cause appreciable differences in the hardness numbers obtained. To avoid errors of this nature some machines, such as the Herbert power-operated machine, are designed to eliminate the personal effect when applying the load.

In practice the hardness number for a given diameter of ball and of impression is found from Tables. The curve in Fig. 111 shows how the Brinell hardness numbers vary with the diameter of impression.

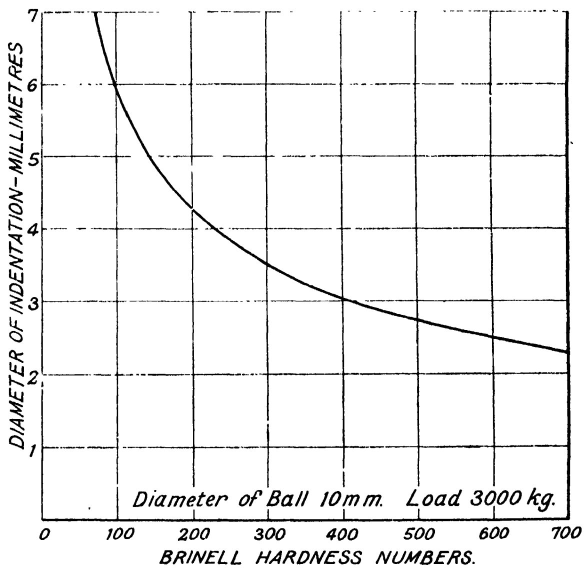


FIG. 111. GRAPH CONNECTING DIAMETER OF INDENTATION WITH HARDNESS NUMBER

Relation Between Brinell Hardness Number and Tensile Strength. The Brinell test is often used to obtain an indication of the strength of a material, since a useful relation exists between the Brinell hardness number and the tensile strength, at least in the case of certain steels. So far, however, no rule of general application has been found to exist.

The chain dotted lines drawn through points represented by circles in Fig. 112 link up the results of a large number of tests made by Messrs. Hadfield and Main on a variety of steels. The British Standard Specification suggests that for steel the tensile strength in tons per square inch can be found approximately by multiplying the Brinell hardness number by 0·22. The full line in the diagram is drawn with this value for its slope. The points marked +, which represent values of

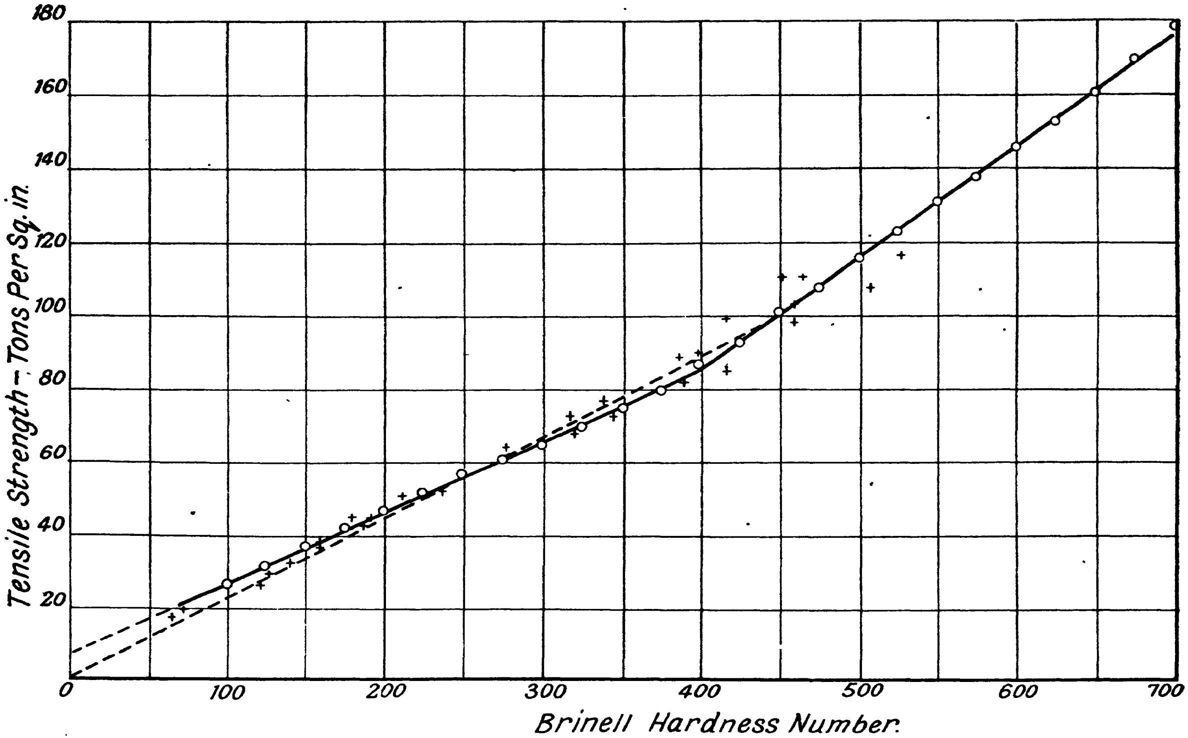


FIG. 112. HARDNESS NUMBER AND TENSILE STRENGTH OF STEEL

the Brinell number, are plotted from various published tests and serve to show how closely the ratio 0·22 holds.

Messrs. Greaves and Jones from an examination of over 1 000 specimens recommend the following values—

Material	Tensile Strength in Tons per in.^{2} = Brinell Number Multiplied by
(*a*) For heat-treated alloy steels with a Brinell number of 250 to 400	0·21
(*b*) For heat-treated carbon steels and for alloy steels with a Brinell number below 250 . . .	0·215
(*c*) For medium carbon steels as rolled, normalized or annealed	0·22
(*d*) For mild steels rolled, normalized or annealed .	0·23

The above values do not apply to severely cold worked or austenitic steels.

For non-ferrous wrought alloys such as duralumin and Y-alloy

$$\text{Tensile strength} = \frac{\text{Brinell hardness number}}{4} - 1.$$

Hardness Testing Machines. Numerous machines for making the Brinell test are now on the market. The loading is generally direct or by means of a lever, by screw gearing or by hydraulic pressure.

Messrs. Alfred Herbert Ltd. make a small machine for determining the hardness of specimens of the order of 0·01 in. in thickness. Amongst the applications of this machine may be cited the testing of thin walled tubes without internal support, and of cutlery blades which would be disfigured by a large impression, the determination of the hardness of wire at successive stages of drawing, and the hardness of a cutting tool close to the edge.

The balls used are 1 or 2 mm. diameter and loads up to 50 kg. are employed. For testing extremely soft material balls of 5 mm. diameter are supplied.

Alfred Herbert Small Hardness Tester. The machine, shown in Fig. 113, is the outcome of investigations made at the Research Department at Woolwich Arsenal with the object of evolving a means of determining accurately the hardness of thin specimens. The table for the reception of the work to be

tested is adjustable by a handwheel and the base is furnished with levelling screws. At the top of the columns is mounted the crosshead, in the centre of which is a fine pitched non-rotating screw. The crosshead can be raised or lowered by a handwheel provided with a thrust bearing. The lower end of the screw carries a suspension stirrup, the upper part of which is attached to the ball-holder while the lower portion carries the weights.

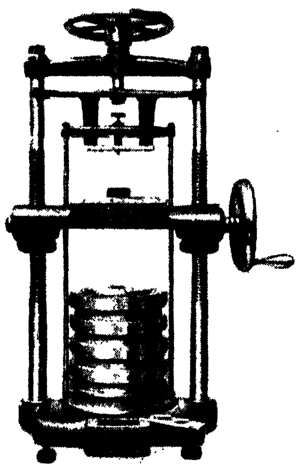


FIG. 113. ALFRED HERBERT SMALL HARDNESS TESTER
(*Alfred Herbert Ltd.*)

The connection between the two stirrups consists of a spherical surface in the loading stirrup bearing in a hole in the other. The loading stirrup supports the suspension stirrup and any pendulum action is effectively damped out by the friction at the spherical surface.

The ball is attached to the ball-holder by indiarubber solution so that the changing of balls is an easy matter.

When the table is at the required height the specimen is placed under the ball. The upper handwheel is then turned until the ball rests on the specimen, the loading stirrup being free and disconnected from the suspension stirrup.

The whole weight is then on the specimen and free from frictional constraint. When the load has been applied for the

prescribed period the upper handwheel is turned back to take the weight off the specimen, which is then removed and the diameter of the impression measured. For this purpose a microscope is supplied, magnifying 180 diameters and with gradua-

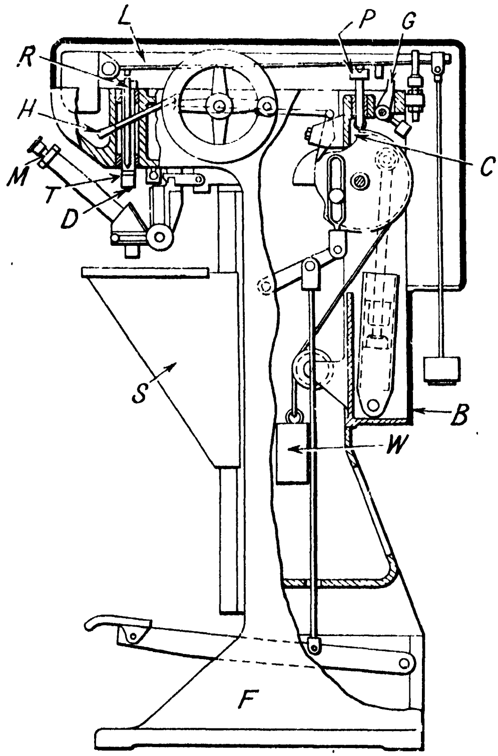


FIG. 114. VICKERS HARDNESS TESTING MACHINE
(*Vickers-Armstrongs Ltd.*)

tions of $\frac{1}{200}$ mm. on the scale. Additional objectives may be obtained giving one-half and one-fifth of the standard magnification respectively.

The Vickers Pyramid Hardness Testing Machine. The Vickers pyramid hardness testing machine, Fig. 114, consists of a frame *F* of U-section which carries the stage *S* and a beam *L* of 20 : 1 ratio. The load is applied through a thrust rod *R* to

the tube T which is free to reciprocate vertically. The tube carries a diamond indenter D at its lower end.

Attached to the main frame is a frame B. This carries the control mechanism. The plunger P is reciprocated by means of the rotating cam C and serves to apply and release the test load. The cam is mounted on a drum and when the starting handle H has been depressed the whole is rotated by a weight W attached by a flexible wire, the speed of rotation being controlled by a piston and oil dashpot.

The rate of displacement of the oil is regulated by an adjustable control valve. The plunger carries a rubber pad at its upper extremity and this engages with a cone mounted in the beam, thereby ensuring a slow and diminishing rate of application for the last portion of the load. The cam both lowers and raises the load, the object aimed at being to eliminate inertia errors. Depression of the foot pedal returns the cam, drum and weight to their original positions.

A trip-piece G supports the beam during this latter operation and drops out as soon as the plunger returns to its top position. The machine is then ready for another test and no external power is required other than that provided by the operator in depressing the foot pedal.

The microscope M is capable of measuring to 0·001 mm. It is mounted on a hinged bracket so that it can be swung to a position immediately over the impression.

Instead of the usual scale or eyepiece micrometer, a specially designed micrometer ocular is used. The impressions are read to knife-edges, thus avoiding eyestrain, and readings are taken entirely from a digit counter mounted on the side of the ocular. In making a test the pedal should first be depressed in order to load the machine. The area to be tested is placed on the stage and the latter raised until the surface to be tested just clears the point of the diamond indenter. The starting handle is then pressed to release the mechanism, when the test proceeds automatically and terminates with an audible click. Measurements are made across the corners of the square impression in the following manner.

The left-hand knife-edge is adjusted by means of a knurled thumb screw to correspond with the left-hand corner of the impression and the right-hand knife-edge, which is controlled by a micrometer screw connected to the counting mechanism, is moved to correspond with the right-hand corner of the

impression. The view through the microscope then appears as in Fig. 115. The reading is taken from the figures on the counter at the side of the eye-piece and converted by means of tables to Vickers pyramid numerals.

In cases where work has to be tested with a view to ascertaining whether or not it conforms to specified maximum and minimum limits of hardness, a third knife-edge is brought into use. This third knife-edge is brought into the field of vision by turning a thumb screw at the side of the ocular and is set by means of the micrometer right-hand knife-edge to correspond with the smaller dimension, i.e. the maximum limit of hardness. The micrometer knife-edge is then adjusted to correspond with the larger dimension, i.e. the minimum limit of hardness.

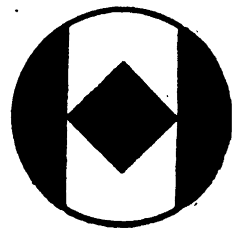
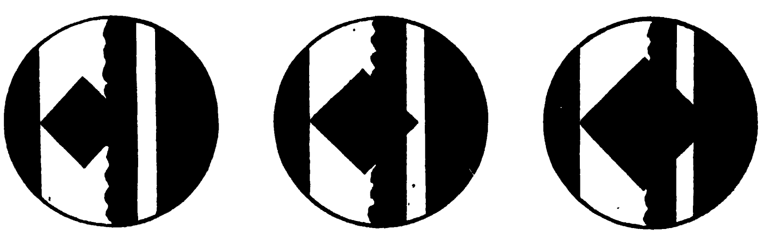


FIG. 115. METHOD OF SETTING THE VICKERS MICROSCOPE
(*Vickers-Armstrongs Ltd.*)

Having made the setting of the second and third knife-edges in this manner it is only necessary, when reading, to set the fixed knife-edge to the left-hand corner of the impression in the ordinary way. Fig. 116 (*a*) shows the material too hard,



(*a*) (*b*) (*c*)

FIG. 116. TESTING FOR CORRECT HARDNESS
(*Vickers-Armstrongs Ltd.*)

Fig. 116 (*b*) shows the material to be correct, and Fig. 116 (*c*) shows that the material is too soft.

In handling mass routine work a convenient method of operation is first to make all the impressions on the pieces, using a jig for locating each piece beneath the diamond. When this has been done the readings are made with the microscope, the

pieces being easily set in position by means of the jig. Up to two hundred tests an hour can be made in this way. The machine is manufactured by Messrs. Vickers-Armstrongs Ltd., Crayford, Kent.

The Olsen-Brinell Hardness Tester. In the Olsen-Brinell hardness tester, Fig. 117, the ball is attached to the underside of a piston to which fluid pressure is applied. The test piece is placed on the table and raised by means of the handwheel until it makes contact with the ball. The piston is a ground fit in its cylinder and without packing, so that when pressure is applied by a small hand pump frictional effects are negligible. The pressure is indicated on the gauge, but to ensure correct loading the plunger carrying the weights seen in the illustration fits into the cylinder above the load piston and ensures that an overload is not applied to the ball.

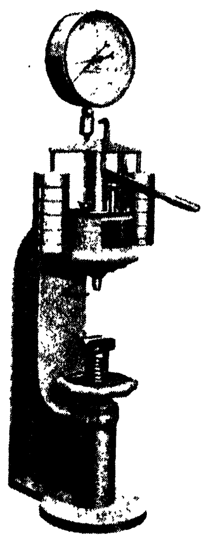


FIG. 117. OLSEN-BRINELL HARDNESS TESTING MACHINE
(*Edward G. Herbert Ltd.*)

When the maximum pressure is reached the plunger and weights "float" and thus limit the load on the piston. Oil leaking past the piston or plunger is drained away to the reservoir. The pressure is released by opening a small valve on the top of the pressure chamber.

Pyramid Hardness Numerals. The advantage of using the pyramid indenter lies in the similarity of the impressions produced. The hardness numerals obtained by using a pyramidal form are independent of the load. This can be seen from the test results plotted in Fig. 118.

The scale of numerals depends on the angularity of the pyramid. The angle selected as a standard in the Vickers system is 136° which, as may be seen from Fig. 119, agrees with the cone angle for a ball impression of 0·375 times the diameter of the ball. Tables of Diamond Pyramid Hardness Numbers are now issued by the British Standards Association (B.S.S.427); the loads specified being 5, 10, 20, 30, 50, 100, and 120 kg. The

numerals obtained with a pyramid of this angularity are identical with those obtained in the Brinell test under appropriate conditions.

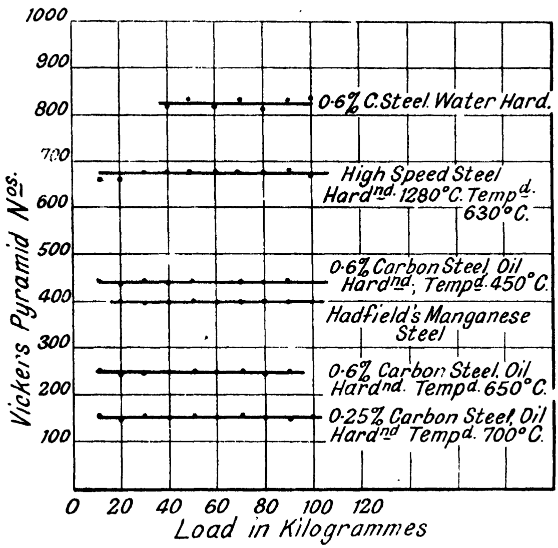


FIG. 118. SHOWING THAT PYRAMID HARDNESS NUMERALS ARE INDEPENDENT OF THE LOAD

The numerals obtained with the Vickers 136° diamond pyramid are termed Vickers pyramid numerals, abbreviated as

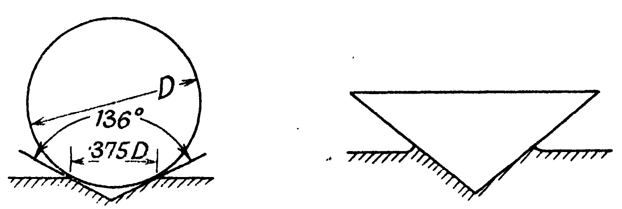


FIG. 119. COMPARISON BETWEEN BALL AND DIAMOND IMPRESSIONS

. . . ° V.P.N. As in the case of the ball the hardness number is defined as the ratio

$$\frac{\text{Load in kg.}}{\text{Surface area of indentation in mm.}^2}$$

If

H = pyramid hardness number.

P = pressure in kg.

d = mean diagonal of impression in mm.

θ = angle between each pair of opposite faces of the pyramid,

then, referring to Fig. 120,

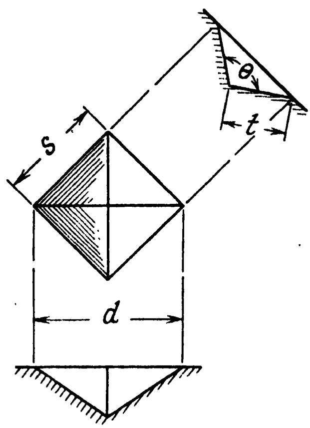


FIG. 120. DIMENSIONS OF DIAMOND IMPRESSION

$$\begin{aligned}\text{Surface area of indentation} &= 4 \times \text{area of one face} \\ &= 4 \times (st/2) \\ &= 4 \times (s/2) \times (s/2) \operatorname{cosec} (\theta/2) \\ &= (d^2/2) \operatorname{cosec} (\theta/2);\end{aligned}$$

therefore

$$H = \frac{2P \sin \theta/2}{d^2}$$

For an angle of 136°

$$H = 1{\cdot}854P/d^2$$

The similarity between the two systems of numerals obtains only in the lower regions of hardness—that is, where the steel ball does not undergo any appreciable deformation. Between

500 and 600 Brinell hardness this deformation begins to make itself manifest by yielding slightly lower readings than the diamond pyramid, and this tendency increases with increasing hardness until it becomes very pronounced. The real relationship is illustrated by curve *A* in Fig. 121. Curve *B* in the same

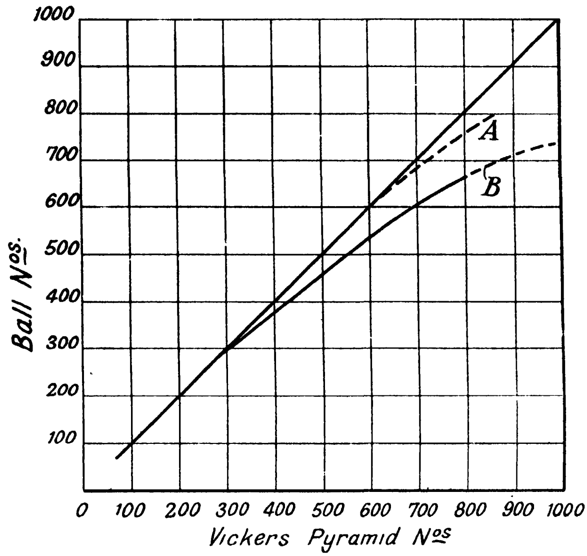


FIG. 121. CURVE SHOWING THE RELATION BETWEEN THE NUMERALS WITH THE VICKERS PYRAMID AND THOSE OBTAINED WITH A STEEL BALL

A loaded to give an impression of which the diameter is equal to 0·375 times the diameter of the ball.
B is a 1-cm. steel ball with a load of 3 000 kg.

figure shows the relation between the hardness numerals obtained with a Vickers pyramid and Brinell numerals obtained in the normal way, that is with 3 000 kg. load and a 10 mm. ball. The disparity is greater here than in the curve A, because in addition to deformation of the ball in the latter case the load is constant. The ball impressions decrease in size with increase in hardness of the material and the hardness numbers are disproportionately low. Unavoidable differences in the hardness of different steel balls will affect the numerals obtained, particularly in the higher regions. On this account the higher portions of the curves are shown dotted.

Rockwell Hardness Tester. Another static hardness tester is the Rockwell machine, Fig. 122, largely used in America. A steel ball $\frac{1}{16}$ in. diameter or a 120° diamond cone with a rounded point is used, and the depth of the indentation is automatically recorded on a dial. The load is applied by the hand screw seen in the figure. To obviate errors caused by the spring of the machine a load of 10 kg. is first applied and the clock indicator

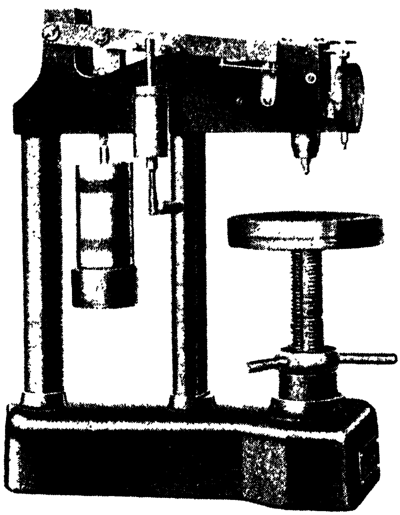


FIG. 122. ROCKWELL HARDNESS TESTER
(*George H. Alexander Ltd.*)

set to zero. The load is then increased to 100 kg. in the case of the ball and to 150 kg. in the case of the diamond. The load is then taken off except for the initial 10 kg. and the hardness numeral read off from the dial. Two scales are employed; one, the "C" scale, for the diamond and the other, the "B" scale, for the steel ball.

In Fig. 123 curves are given for the approximate conversion of Rockwell's "B" scale to Brinell hardness numbers. The machine is rapid in action and articles may be tested at the rate of 250 per hour.

Shore Scleroscope. A dynamic hardness test is provided by

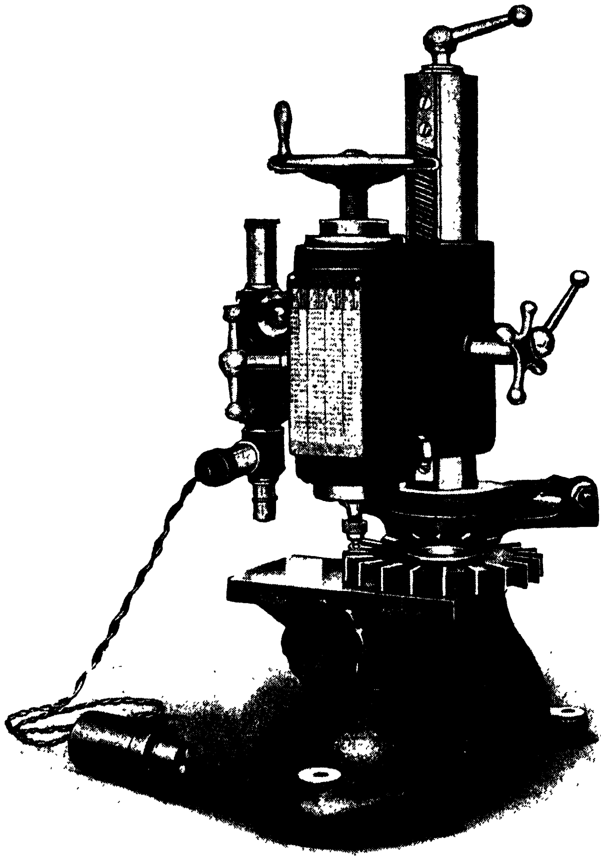


FIG. 123. FIRTH HARDOMETER
(Thomas Firth & John Brown, Ltd.)

the Shore scleroscope in which a small pointed tup weighing about 0·0052 lb. is allowed to fall freely from a height of 10 in. on to the test piece.

The height of rebound of the tup is measured against a scale graduated into 140 equal divisions. The result is dependent on the permanent deformation produced in the test piece at the point of impact, the rebound being diminished on account of the work of deformation.

As most materials are hardened by cold working it is important when making tests to see that the material is not struck twice in the same spot.

The Firth Hardometer. The Firth Hardometer (Fig. 123), made by Messrs. John Brown & Thomas Firth, Ltd., is a spring-loaded machine; this method being employed in order to reduce inertia effects. The spring is operated at a comparatively low stress and retains the constancy of its elastic properties over long periods.

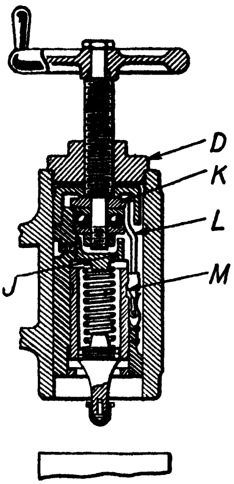


FIG. 124. FIRTH HARDOMETER: LOADING MECHANISM
(Thomas Firth & John Brown, Ltd.)

A microscope is used for measuring the indentations. It is claimed that the magnification of the scale of the microscope provides a more reliable method of measurement than the use of a mechanical depth measuring device which is called upon to function on each reading made and which, at the same time, must remain sufficiently sensitive to deal with exceedingly small impressions. Either a ball or a diamond indenter may be used with the machine.

The arrangement of the loading mechanism is shown in Fig. 124. As the handwheel is turned, the load cylinder to which the ball holder is attached is forced downwards through the medium of its spring. The load cylinder carries with it the trigger *M* which engages the recess in the inner bushing. This allows the stop *L* to engage the ratchet wheel *K* as soon as the correct load has been applied. Provision is made for calibration, but the makers recommend the return of the machine for this purpose.

The machine is intended for fixed loading and is made in capacities of 120 kg., 30 kg., and 10 kg., to cover a wide range of materials and thicknesses. The smaller size is intended mainly for use with a diamond indenter, but a 1 mm. ball may also be used.

For diamond impressions the British Standards Institution recommends a minimum thickness of test piece of 10·5 times the depth of impression, while for ball impressions the minimum thickness should be 7 times the depth of impression.

The following table is based on these recommendations—

Load (kg.)	Indenter	Depth of Impression (inches)	Minimum Thickness of Material for Test (inches)
120	2 mm. ball	0·725/H	5·264/H
	4 mm. ball	0·376/H	2·632/H
	Diamond	0·0840/H	0·882/H
30	1 mm. ball	0·376/H	2·632/H
	2 mm. ball	0·188/H	1·316/H
	Diamond	0·0420/H	0·441/H
10	Diamond	0·0242/H	0·254/H

H is the hardness of the material as determined by the test used.

As a rough guide to the approximate minimum thickness of a few of the common metals which may be tested using the 30 kg. Hardometer with a 2 mm. ball, the following table may be of service—

Metal	Brinell Hardness Number	Minimum Thickness (inches)
Copper	48	0·027
Nickel Silver (cast)	61	0·021
Nickel Silver (as rolled)	70	0·018
Manganese Bronze (cold rolled)	106	0·012
Duralumin	115	0·011
Aluminium Bronze (cold rolled)	119	0·011
Phosphor Bronze (cold rolled)	120	0·011
Brass (cold rolled)	124	0·0105
Monel Metal (forged)	167	0·008

Relations Between the Various Systems of Hardness Numbers. No precise relationship exists between scleroscope, Rockwell and Brinell numbers. The Brinell number is of the order of 6 times the scleroscope number. Comparisons between the various systems of numerals are given in Figs. 125 and 126.

Experiments on copper alloys show that when these are subjected to mechanical treatment which causes a change in the hardness of the material an approximate straight-line relationship exists between scleroscope and Brinell numbers and between Brinell numbers and the reciprocals of the Rockwell numbers.

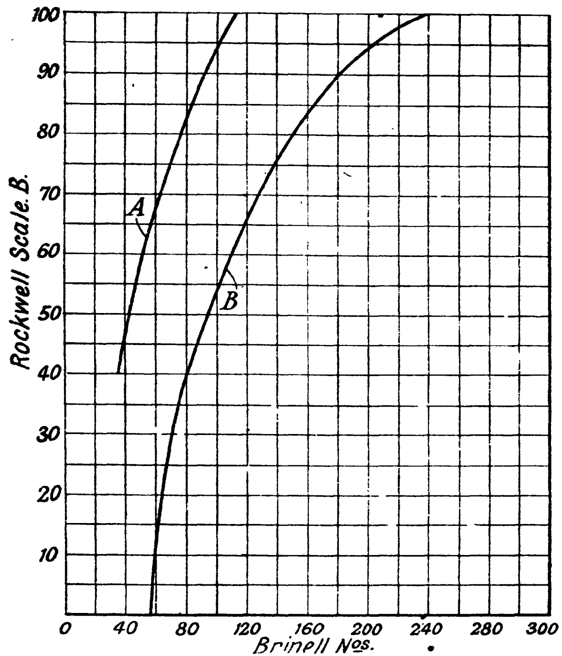


Fig. 125. Conversion to Standard Brinell Numbers of Rockwell B. Scales

(*a*) Rockwell B scale $\frac{1}{16}$ in. ball and 60 kg. load.
(*b*) Rockwell B scale $\frac{1}{16}$ in. ball and 100 kg. load.

(*Institution of Automobile Engineers*)

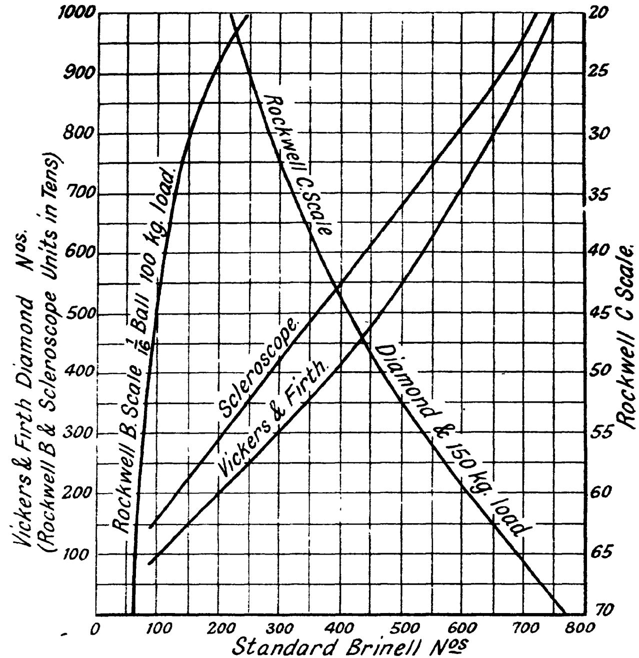


Fig. 126. Conversion Chart

Rockwell. Scleroscope, Vickers and Firth diamond numbers to Brinell numbers

(*Institution of Automobile Engineers*)

It is, however, necessary to determine the constants for each particular alloy.

Distortion in Diamond Impressions. The ridge effect found in ball impressions has its counterpart in diamond impressions in the shape of a slight convexity or concavity of the edges of the square impressions, Fig. 127. This effect has been examined by O'Neill, who finds that the concavity is due to a raised extruded ridge formed along the faces of the pyramid, while convexity is due to a depression of the edge. Soft copper, soft iron and quenched steel show a slight apparent convexity, while drawn copper and rolled steel appear to give concave indentations. The effect introduces an error into the measurements whereby worked metals will give high results and very workable metals will give low results.

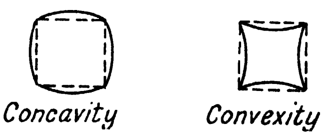


Fig. 127. Ridge Effects in Diamond Impressions

Herbert Pendulum Hardness Tester. Another apparatus for making dynamical hardness tests is the pendulum hardness tester of Messrs. Edward G. Herbert Ltd. It consists, as its name implies, of a pendulum weighing 4 kg. supported on a ball 1 mm. diameter or on a 1 mm. ball-shaped diamond, the whole constituting a compound pendulum of $\frac{1}{10}$ mm. length. Immediately above the ball is a graduated weight mounted on a screw. By raising or lowering the weight the centre of gravity of the instrument may be brought to a predetermined distance below the centre of the ball. Above the weight is a curved spirit level and a scale reading to 100. The pendulum is shown mounted on an operating stand in Fig. 128.

Six practical tests can be made with the instrument:

(1) A Time Hardness Test. When the pendulum is allowed to rest on the specimen the ball makes an indentation the size of which depends on the hardness of the specimen. The pendulum is allowed to oscillate through a small arc and the time of swing noted. The time of swing, measured by a stop watch, gives a measure of the hardness.

Microscope readings are not needed. The method of testing eliminates distortion of the ball which occurs when measurements of Brinell hardness above 600 are being made, and the use of a ball-shaped diamond enables tests on steels of 1 000 Brinell hardness to be made.

TABLE VIII

TYPICAL HARDNESS NUMBERS

Using a 1 mm. steel ball

Glass	100
Very hard carbon steel	75
Hard carbon steel	65
Heat-treated h.s. steel	52
Annealed h.s. steel	26
Mild steel	20
Rolled brass	15
Cast brass (soft)	11
Lead	3

The approximate Brinell hardness may be obtained by multiplying diamond time hardness by 13·5 and steel ball time hardness by 10.

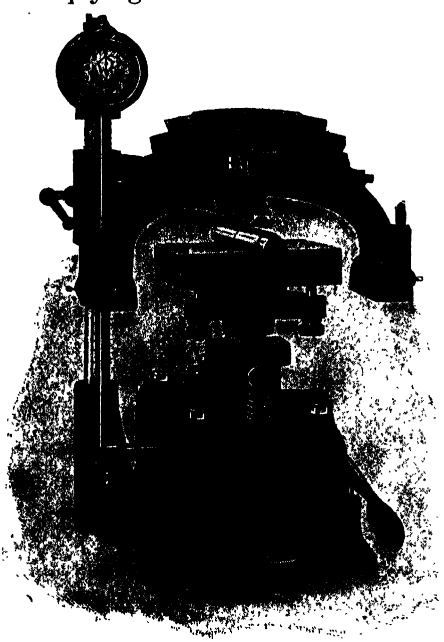


FIG. 128. HERBERT PENDULUM HARDNESS TESTER

(Edward G. Herbert Ltd.)

(2) TIME-WORK-HARDENING TESTS. Metals in service become work-hardened in varying degrees, and a knowledge of their work-hardening properties is of importance; as for example in metals for deep drawing or pressing, free cutting steels, and rail, tyre and gear steels, in which work-hardening increases their resistance to wear.

To make the time-work-hardening test the time-hardness test (1) is first made and the specimen is then work-hardened by rolling it with the pendulum. A second time-hardening test is then made. The processes are repeated alternately and continued until the hardness reaches a maximum (the "maximum induced hardness") and then declines.

Some representative results are given below.

TABLE IX

TIME-WORK-HARDENING TESTS

Material	Original Time Hardness	Passes of Ball					
	0	2	4	6	8	10	12
Hard tool steel	71	89	88	86	—	—	—
Manganese steel	21	45·5	52·4	54	56·2	57·2	44·6
Mild steel	21	30	30·8	30·9	31·2	32·3	31·6
Stainless steel	18	38·7	41·9	43·1	43·1	44	43·8
Loco. tyre steel	28·6	35	36·2	36	—	—	—
Deep drawing steel	12	19·4	20·7	20·6	—	—	—

(3) SCALE TEST. If the pendulum be tilted and placed on a specimen with the bubble at O on the scale it will indent the specimen as before. When released it will swing and elongate the impression by rolling, the length of swing indicating the resistance of the materials to rolling.

The readings obtained in this test generally place materials in the same order of hardness as the time test but the "scale-hardness" numbers do not correspond with the time-hardness numbers. The scale test is very sensitive and is chiefly used in detecting changes in hardness, especially those caused by heating or working the metal.

(4) HOT HARDNESS TEST. Specimens can be subjected to a time-hardness test with the specimen in an electric furnace, the temperature being measured by pyrometer. A ball-shaped diamond is necessary in this test as a steel ball would be affected by the high temperature.

(5) TEMPERATURE-WORK-HARDENING TEST. Work-hardening tests on unhardened metals have shown that these usually lose their work-hardening capacities at comparatively low temperatures and regain them at higher temperatures, although the indentation hardness remains nearly constant.

A typical work-hardening curve of Vickers test bar steel is given in Fig. 129. Three peaks, P_1, P_2, P_3 and three depressions D_1, D_2, D_3 are frequently present, the principal depression D_1 occurring at 120° to 140° C. in steels and below 100° C. in other metals.

(6) Scale-Work-Hardening Test. If the scale test has been made as described in (3) and at the end of the first swing the pendulum is tilted to 100 and then released, the reading on the reversal will usually be higher, as the specimen has been work-hardened by the ball. If this is repeated from 0 to 100 alternately, the succession of readings obtained will show the progressive increase of hardness due to working.

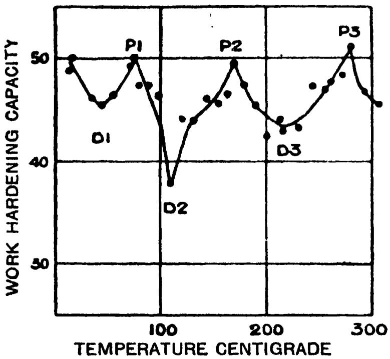


Fig. 129. Work Hardening Curve Obtained with the Pendulum Hardness Tester
(*Edward G. Herbert Ltd.*)

The work-hardening capacity of a metal is measured by the average increase of scale hardness caused by rolling four times with the pendulum ball.

Theory of the Pendulum Hardness Tester. It has been suggested by E. G. Herbert that three simple types of hardness are involved, namely, *plastic indentation hardness* (Brinell hardness); *elastic indentation hardness*—a measure of which is given by the pendulum-time-test on highly elastic materials; and *flow hardness* as measured by the scale-time ratio. Dr. W. J. Walker, however, suggests that only two material properties are involved, *viscous* or *plastic hardness* and *elastic hysteresis hardness*.

Timoshenko has put forward the following simple theory of the instrument.

Let e, Fig. 130, be the distance of the c.g. (G) of the apparatus from the centre O of the ball, W the weight of the pendulum, I its moment

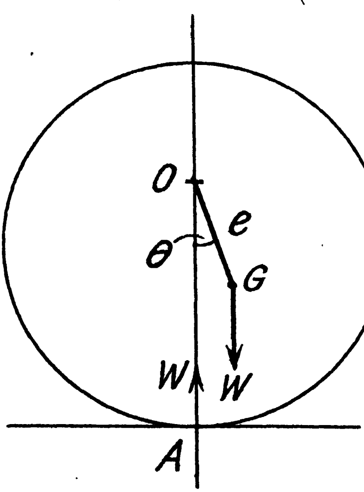


Fig. 130. Illustrating Timoshenko's Theory of the Pendulum Hardness Tester

of inertia about an axis through O perpendicular to the plane of oscillation.

Assuming e and θ to be small quantities the equation of motion when the apparatus is supported on a rigid plane surface is

$$I(d^2\theta/dt^2) + eW\theta = 0$$

from which the periodic time of one swing, that is, one half of a complete oscillation, is

$$T = \pi\sqrt{(I/geW)}.$$

This result is assumed to apply in the case of an elastic plane surface as the distribution of pressure over the area of contact will be symmetrical about the vertical axis.

When permanent set occurs, Fig. 131, the distribution of pressure is no longer symmetrical about the vertical axis OA. Timoshenko assumes the length of the arm of the couple to vary as $(e + \delta)$ where δ is a constant depending on the permanent set, so that the time of one swing is now

$$T = \pi\sqrt{[I/g(e + \delta)W]}$$

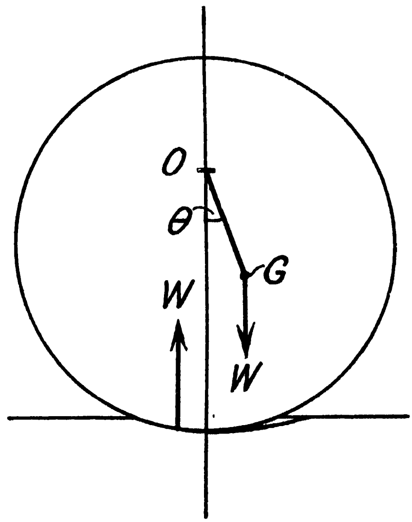


Fig. 131. Illustrating Timoshenko's Theory of the Pendulum Hardness Tester

In a comparison of calculated and experimental results on glass and high-speed steel the latter formula gave consistent results.

The first formula shows the time of oscillation to be inversely proportional to $\sqrt{e}$ and thus independent of the hardness qualities of the material. For some comments on this theory and for some experimental results see *The Engineer*, 1923 (6th April, 6th July, and 7th September).

CHAPTER IX

NOTCHED-BAR IMPACT TESTING

Impact Tests. In a great many cases in practice the loads on machine members are more or less suddenly applied and hence arises the need for a study of the effects produced by dynamic loading. Static tests, while furnishing valuable information, are nevertheless insufficient to bring out all the characteristics that bear on service conditions.

The point may be illustrated by a test by Philpot on nickel-chrome steel.

Specimen	Tensile Test (Static)					Notched-bar Test (Dynamic)
	L.P.	Y.P.	Ult. Stress	Elongation %	R.A. %	Impact Value
	tons per sq. in.					*ft.-lb.*
(1)	40·4	47·8	55·5	28·6	64·0	78
(2)	39·6	45·9	54·3	26·5	63·7	9·1

From the results of the tensile test it might be inferred that the two specimens are of identical structure, but from the figures in the last column, which represent the energy absorbed —the impact value—when the test pieces in the form of notched bars are subjected to a hammer blow, it is seen that, in some respect, a wide divergence exists. This divergence is to be accounted for by the effect of different heat treatments received by the specimens prior to testing. The treatment left unaffected their mechanical properties as diagnosed by the tensile test but left one material much less able than the other to sustain a shock of the type imposed. Specimen (1) had received the correct heat treatment; specimen (2) had not.

The mechanical properties exhibited by a material under a load suddenly applied depend on the suddenness of application. An extreme instance is that of a mild steel bar, a ductile material which, subjected to a shock wave produced by a sufficient charge of gun-cotton, suffers a characteristically brittle fracture. Much less severe conditions will produce a

similar result. The subject is of considerable importance and keen interest is now being taken in high-velocity testing as distinct from the theme of this chapter, the notched-bar impact test. In this, a dynamic test, a specimen in the form of a notched-bar is tested as a beam under impact from a hammer or pendulum. The term "impact" in this connection is somewhat unhappily chosen as the test indicates not so much the resistance of the material to shock or impact but rather differences of condition in the material which are brought to light through the concentration of stress which occurs at the notch and which other tests fail to demonstrate. It may be remarked that in the British Standard Specification of the notched-bar test the term "impact" does not appear. The notch localizes the stress and determines the behaviour of the material.

The chief value of the test lies in indicating whether or not the heat treatment of a steel has been carried out in a satisfactory manner. It is generally conceded that the condition of a steel giving a high impact value is better than that of the same steel which gives a low impact value. Conclusive evidence on this point appears to be lacking, but the view is supported by the results of experience.

The notched-bar test figures prominently in Air Board Specifications and among the reasons given for its inclusion are—

(*a*) That it is the test which gives most information as to whether the heat treatment of a steel is satisfactory or otherwise.

(*b*) That if the steel has been heat treated to the best advantage the impact value will be relatively high, but if the heat treatment has not been satisfactorily carried out the impact value will be relatively low.

The impact value, however, must not be judged by itself but must be considered after taking account of the type of steel upon which the result was obtained.

Of machines for making impact tests the best known are the Frémont, Charpy, Izod, and Guillery, the last being a rotary-disc machine.

Particulars of these are given in Table X.

The Charpy Machine. The Principle of the Charpy machine is illustrated in Fig. 132 (*a*). The hammer or pendulum *H* swings, on ball bearings and is in the form of a disc provided with a vertical knife *K*. The specimen being simply supported as

TABLE X

DIMENSIONS OF IMPACT MACHINES

Machine	Type	Height of Fall		Distance between Supports		Velocity of Impact		Striking Energy	
		m.	ft.	mm.	in.	m. per sec.	ft. per sec.	kg.-m.	ft.-lb.
Frémont	Falling tup.	4·0	13·1	22	0·865	8·86	29	—	—
Charpy (large)	Pendulum	3·144	10·3	120	4·73	9·68	32	300	2 170
Charpy (small)	,,	1·33	4·36	40	1·58	5·53	17·33	30	217
Izod (large).	,,	0·61	2·0	22	0·865	3·5	11·4	16·6	120
Izod (small)	,,	—	—	—	—	3·05	10	3·2	23
Guillery	Rotating disc	—	—	40	1·58	8·86	29	60	434
Hounsfield Balanced Impact . .	Pendulum	—	—	38·1	1·5	6·92	22·7	6·6	48

shown at (*b*), Fig. 132, the pendulum is raised to the position indicated by the dotted lines, by means of a worm gear, when it is released and allowed to fall and fracture the test piece.

In its upward swing the pendulum carries the pointer *P* over the semi-circular scale *S*, graduated in degrees. Corresponding to the angle through which the pointer is carried, the energy absorbed in breaking the specimen is read from a table.

The tapped hole at the back of the disc is intended to receive specimens for tension-impact tests; a special attachment is provided for the purpose. This test is not standardized.

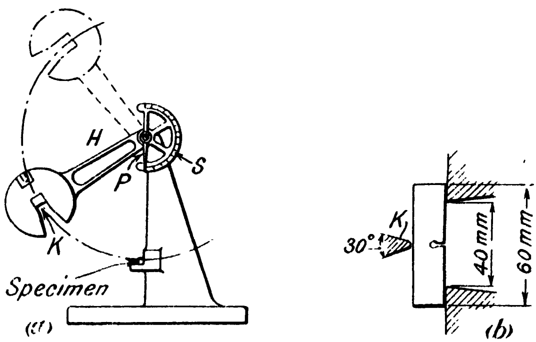


FIG. 132. CHARPY IMPACT TESTING MACHINE
(*a*) Principle of machine. (*b*) Setting of test piece.

The striking velocity of the small Charpy machine is 5·28 metres per sec., i.e. 17 ft. 4 in. per sec.

The Izod Machine. The Izod machine (Fig. 133) is similar in principle to the Charpy, but the specimen is tested as a cantilever. The knife of the hammer is horizontal and strikes the specimen at a point 22 mm. above the plane of fixing (Fig. 134). The test piece is gripped by a vice and a gauge is provided to aid in setting the test piece correctly in position. The energy absorbed in fracture is read off directly from a scale. The striking velocity is 11·4 ft. per sec.

The 30 kg.-m. Charpy and the Izod 120 ft.-lb. machines both take a specimen 10×10 mm. in section. Some experimenters prefer the Charpy machine on account of the test piece not being stressed in the region of the notch by the grip of a vice. However, the Izod machine is the one adopted in this country for standard tests, though the Charpy finds considerable use.

Both types are used in America but on the Continent the use of the Charpy is practically universal.

The Amsler Machine. A machine designed for making both Izod and Charpy tests is made by Messrs. Amsler of Schaffhouse. The upward swing of the hammer causes a cursor to travel over a vertical scale which indicates directly the energy absorbed in fracturing the test piece.

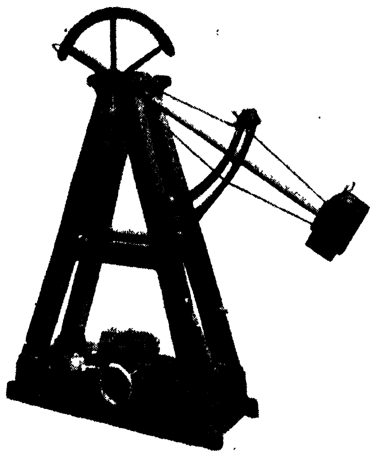


FIG. 133. IZOD IMPACT MACHINE

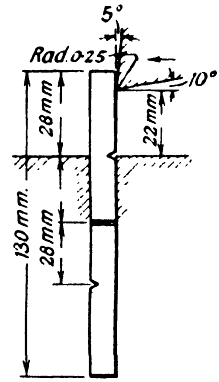


FIG. 134. METHOD OF GRIPPING AND STRIKING TEST PIECE IN IZOD MACHINE

A rope having one end attached to the hammer and the other end wrapped round a spindle on which it can slip under the action of a weight prevents the hammer from falling back after its upward swing. The brake is so arranged that it has no effect on the upward swing of the pendulum.

In making a test the hammer is raised until it engages with a hook on the suspension frame. The hook is disengaged by hand when it is desired to release the hammer.

The striking edge of the hammer is fixed at the centre of percussion so that there is practically no vibration of the pendulum when the bar is struck and no energy absorbed by it.

The striking edge has an included angle of 45° and is rounded to a radius of 3 mm.

A calibrating device is fitted to permit the accuracy of the indicator to be checked.

An additional hammer is supplied for attachment to the pendulum when it is desired to make Izod tests, as well as a vice for gripping the specimen.

The Amsler machine can be arranged for shock tests on wires and strips of sheet metal. For such tests the hammer is provided with two gripping heads in which the ends of the wire or strip are held by wedges with serrated faces. The wire is fixed in the gripping heads and the grips are then slipped into position. One of the gripping heads rests in transverse slots in the hammer while the other is supported freely at the rear of the hammer, but without putting tension on the wire. When the hammer is allowed to fall, projecting wings come against stops in the baseplate and are arrested whilst the hammer passes on and pulls the wire. A somewhat similar device enables tests to be made on shouldered metal bars.

The Hounsfield Balanced Impact Machine. The Hounsfield balanced impact machine employs two moving tups in place of the usual tup and anvil, the total weight being only 39 lb.

The tups are clearly shown in Fig. 135. The supports for the bearings and the release mechanism are carried in a light aluminium frame *F* which can be secured to any table or bench with wood screws, the necessity for a concrete foundation being thereby avoided.

The equivalent mass at the centre of percussion of each tup is 12 lb., and as this falls through 2 ft. the energy stored is 48 ft.-lb., while the relative velocity is 22 ft. per sec.

The tups are held up by catches and are released by throwing over the hammer. When the tups pass one another a non-return pawl mechanism is brought into action which prevents the tups from swinging backwards. As the indicating pointers *P* record the difference in movement between the tups the scale is a very open one.

The test piece is carried by the inner tup in which it is inserted by withdrawing the notch register. The notch register is a spring-operated plunger, with a chisel-shaped end, which engages with the notch of the test piece and is exactly opposite the knife edge which supports the centre of the test piece when the blow occurs.

The form of the test piece is shown in Fig. 136. Results

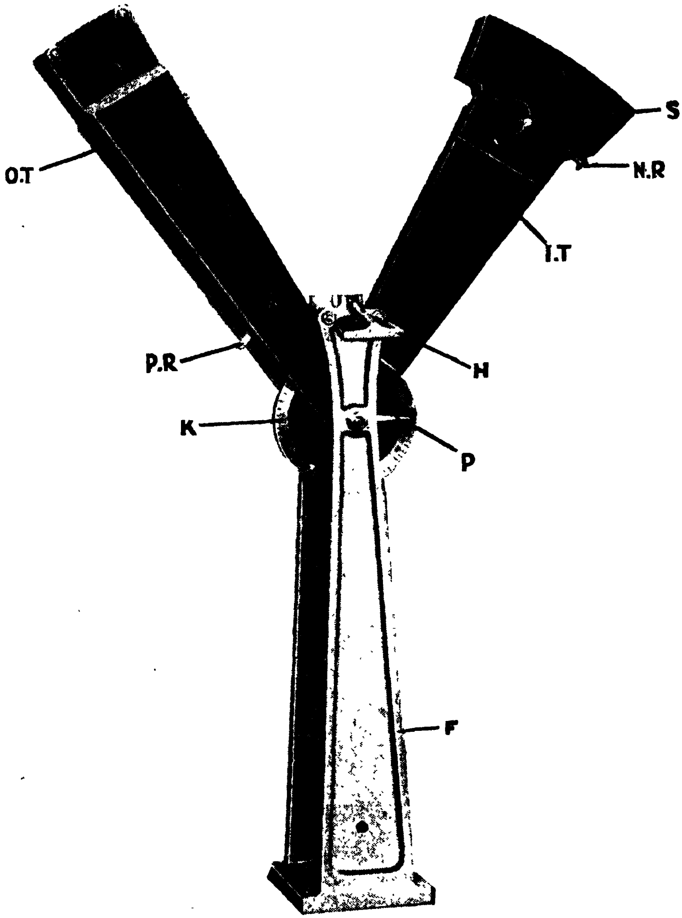


FIG. 135. HOUNSFIELD BALANCED IMPACT MACHINE

F = Frame
H = Hammer
IT = Inside tup
K = Scale
NR = Notch register
OT = Outside tup
P = Pointer
PR = Pawl release lever
S = Test piece

(Tensometer Ltd.)

obtained on the Hounsfield machine, when multiplied by 2·5, give good Izod values.

Theory of the Pendulum Impact Test. The energy absorbed in breaking the test piece is equal to the difference between the initial and final amounts of energy in the hammer, less a

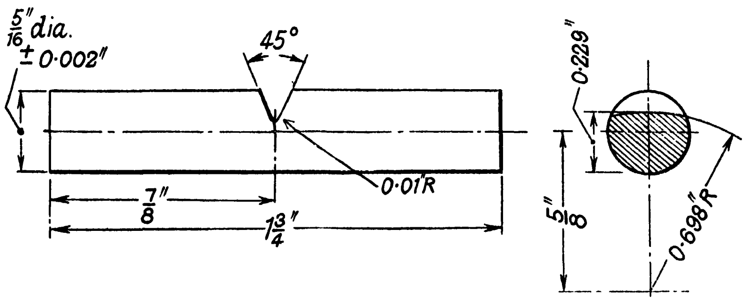


FIG. 136. TEST PIECE FOR USE IN HOUNSFIELD BALANCED IMPACT MACHINE

small amount absorbed in friction and in moving the broken test piece. Referring to Fig. 137, let

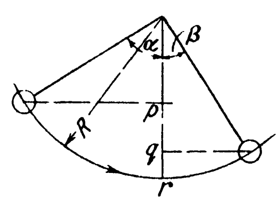


FIG. 137. ILLUSTRATING THE THEORY OF THE IMPACT TEST

α = initial angle of inclination of the pendulum.

β = the angle of rise after fracture.

ϕ = the angle of rise corresponding to α with no test piece in place.

θ = angle of descent needed to give an angle of rise with no test piece in place.

R = radius from the axis of rotation to the c.g. of the pendulum.

K = radius of gyration about the axis of suspension.

W = weight of pendulum.

w = weight of test piece.

Then

$$\text{Initial energy} = WR(1 - \cos \alpha)$$
$$\text{Final energy} = WR(1 - \cos \beta)$$
$$\text{Friction loss on downward swing} = (WR/2)(\cos \phi - \cos \alpha)$$
$$\text{Friction loss on upward swing} = (WR/2)(\cos \beta - \cos \theta)$$

The energy in pendulum just before impact

$$= WR(1 - \cos \alpha) - (WR/2)(\cos \phi - \cos \alpha)$$
$$= (WR/2)(2 - \cos \alpha - \cos \phi)$$

The energy in pendulum just after fracture

$$= WR(1 - \cos \beta) + (WR/2)(\cos \beta - \cos \theta)$$
$$= (WR/2)(2 - \cos \beta - \cos \theta)$$

If ω is the angular velocity after fracture the kinetic energy of the pendulum $= WK^2\omega^2/2g = WV^2/2g$

and $$(V^2/2g) = (R/2)\ (2 - \cos \beta - \cos \theta)$$

Assuming the test piece to move with the velocity V, its kinetic energy is

$$wV^2/2g = (wR/2)(2 - \cos \beta - \cos \theta)$$

Hence, energy absorbed in fracture

$$= (WR/2)(2 - \cos \alpha - \cos \phi) - (WR/2)(2 - \cos \beta - \cos \theta) - (wR/2)(2 - \cos \beta - \cos \theta)$$
$$= (WR/2)(\cos \beta + \cos \theta - \cos \alpha - \cos \phi) - (wR/2)(2 - \cos \beta - \cos \theta)$$

With no friction this becomes

$$WR(\cos \beta - \cos \alpha) - wR(1 - \cos \beta)$$

and neglecting the kinetic energy of the test piece,

$$WR(\cos \beta - \cos \alpha)$$

In practice a table of impact values is used which gives the energy absorbed corresponding to any angle of rise β, or a specially calibrated scale is provided.

If it is desired to calibrate the machine this may be done by releasing the pendulum from an angle of inclination $\phi \gtreqless \alpha$ and noting the angle of ascent β_1. The pendulum is allowed to make another forward swing freely and the angle of ascent β_2

again noted. The mean energy absorbed in an upward swing is $(WR/4)(\cos \beta_2 - \cos \beta_1)$ and represents the friction loss for an angle of ascent $(\beta_1 + \beta_2)/2$.

A repetition of the process enables a curve to be plotted to give the friction loss for any angle of rise.

Another method is to measure the angle of inclination θ of the pendulum before release by means of a protractor of the Starrett or similar type and to note the corresponding angle of rise β.

The loss by friction for the upward swing is then

$$(WR/2)(\cos \beta - \cos \theta)$$

As in actual testing the angle α is constant, the friction loss on the downward swing will be constant and can be determined by the method already described. To determine the constants of the pendulum accurately necessitates the dismantling of the machine, but the product WR of the above formula may be checked by supporting the pendulum in a horizontal position by a spring balance attached to the hammer.

The Guillery Machine. The Guillery machine consists of a totally enclosed steel flywheel rotated by hand. The speed of rotation is indicated by the height of a coloured liquid in a vertical glass tube. When the desired speed has been obtained, indicated by the liquid tachometer as corresponding to a rim speed of 29 ft. per sec., a sliding knife in the rim of the wheel is released. The knife strikes and fractures the test piece which is held in a die at the side of the machine. The speed of the wheel falls by an amount depending on the energy absorbed in fracturing the test piece, and this is indicated by a fall in the level of the liquid in the tube. The tube is calibrated so that the amount of energy can be read off directly.

Influence of the Shape of the Notch. Doubt has been thrown on the impact test as being incapable of giving consistent results, but it has been established that variations in results with a given material are due, not to the mode of testing, but to the lack of homogeneity in the material tested.

The efficacy of the notch in distinguishing the effect of heat treatment is shown by an example given by Sir Robert Hadfield.

A steel of composition

C	Si	Mn	S	P	
0·12	0·02	0·28	0·02	0·02	per cent

after quenching and tempering possessed the following mechanical properties—

Elastic limit . .	16 tons per in.^{2}
Tenacity . . .	28 tons per in.^{2}
Elongation . .	35 per cent
Reduction of area .	65 per cent

Under the Frémont test the specimen bent double cold.

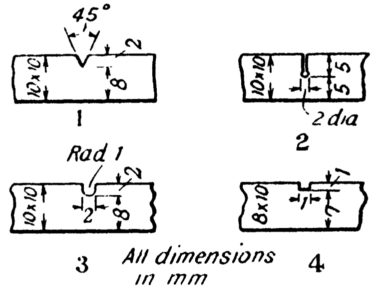


FIG. 138. TYPES OF NOTCH

Portions of the same steel were heated to about 1 200° C. and allowed to cool slowly.

The results then obtained were

Elastic limit . .	9 tons per in.^{2}
Tenacity . . .	22 tons per in.^{2}
Elongation . .	46 per cent
Reduction of area .	64 per cent

Under the impact test this steel was found to be extremely brittle; it snapped almost like cast iron and did not bend 1°.

So far as the notch itself is concerned, any form can be used independently of the type of machine, though not necessarily with advantage.

The standard V-notch 2 mm. deep and 0·25 mm. radius at the root is more economically produced than the Charpy standard having a drilled hole, and can be easily formed by a suitable milling cutter. In either form care should be taken to avoid cross grooves. Several types of notch are shown in Fig. 138, namely (1) Izod, (2) Charpy, (3) Mésnager and (4) Frémont. The standard notch adopted by the British Standards Institution is the Izod 2 mm. deep, 45° included angle with a root radius of 0·25 mm. The standard Izod specimen is shown in Fig. 139. The figure also shows a round form with a straight notch. A curved notch is also used (B.S.S. No. 131). As it is customary to make at least three tests on a sample, the Izod

test piece is often provided with three notches arranged so that tests can be made in different directions relative to the cross-section.

Opinions differ as to the merit of respective notches.

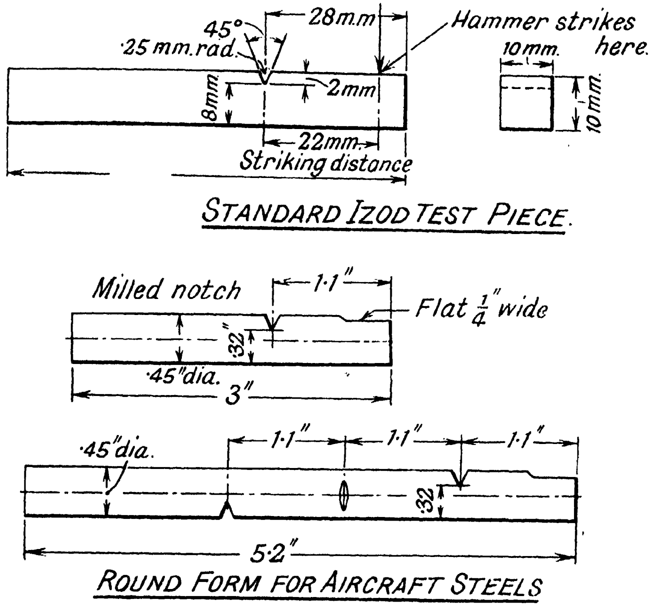


FIG. 139. STANDARD IZOD TEST PIECES

Messrs. Greaves and Moore state that considerable rounding of the notch, as in the Mésnager form, seriously reduces its capacity for distinguishing between tough and brittle material.

According to Petrenko, Bureau of Standards Technological Paper No. 289, the Izod and Mésnager notches are about equally efficient, while the Charpy form shows less difference in the notched-bar characteristics of materials. The results of other work appear to conflict with this but the precise conditions under which tests were made need to be known before conclusions can be fairly drawn. The Charpy notch gives larger values than the Izod for brittle materials, but for tough materials it gives smaller values owing to the reduced thickness of specimen. The Izod notch is stated to be preferable for brittle materials.

A point of supreme importance in notched-bar testing is the relation between test results obtained for the same materials on different machines. Unfortunately, in the present state of knowledge, not only can no general relation be given which will cover all classes of materials, but even the results of tests of the same material on machines of different types are sufficiently discordant to preclude the formulation of very definite laws. The relations given in the sequel must therefore be regarded as approximations which are of service over limited ranges when used with circumspection.

Tests by Philpot on nickel-chrome steels in an attempt to discover an easily- and cheaply-made test piece which would be suitable for both Izod and Charpy machines led to a comparison being made of several notches. The departure from equality of impact values when tests are made of several notches with various steels in the two machines is shown in the plot in Fig. 140.

Another example is given in Fig. 141. In this case the object of the tests was primarily to discover if brittleness existed in the region of a weld in a welded mild steel plate. The numbered rectangles indicate positions from which test pieces were cut. Three types of notch were employed. Particulars of these and of the impact values are given in the figure which is self-explanatory. It will be observed that with the 1·3 mm. drilled notch the Izod and Charpy values are in fair agreement, whereas with the other types the Charpy values exceed the Izod values by some 40–50 ft.-lb.

The curve in Fig. 142 shows the relation between the Mésnager form tested in the Guillery machine and the standard Izod form in the 120 ft.-lb. Izod machine.

Between 10 and 60 ft.-lb. Izod the relation is

$$\text{Guillery} = 0{\cdot}17 \text{ Izod (ft.-lb.)} + 1$$

for steels of the following types—

Percentage Composition							Yield Point Tons per in.²	Max. Load Tons per in.²
C	Si	Mn	S	P	Ni	Cr		
0·3 to 0·4	0·15	0·65	0·04	0·04	—	—	19 to 24	34 to 44
0·3 to 0·4	0·15	0·6	0·04	0·04	3·5	—	27 to 36	42 to 55
0·3 to 0·4	0·15	0·4	0·04	0·04	3·5	0·6	30 to 40	42 to 52

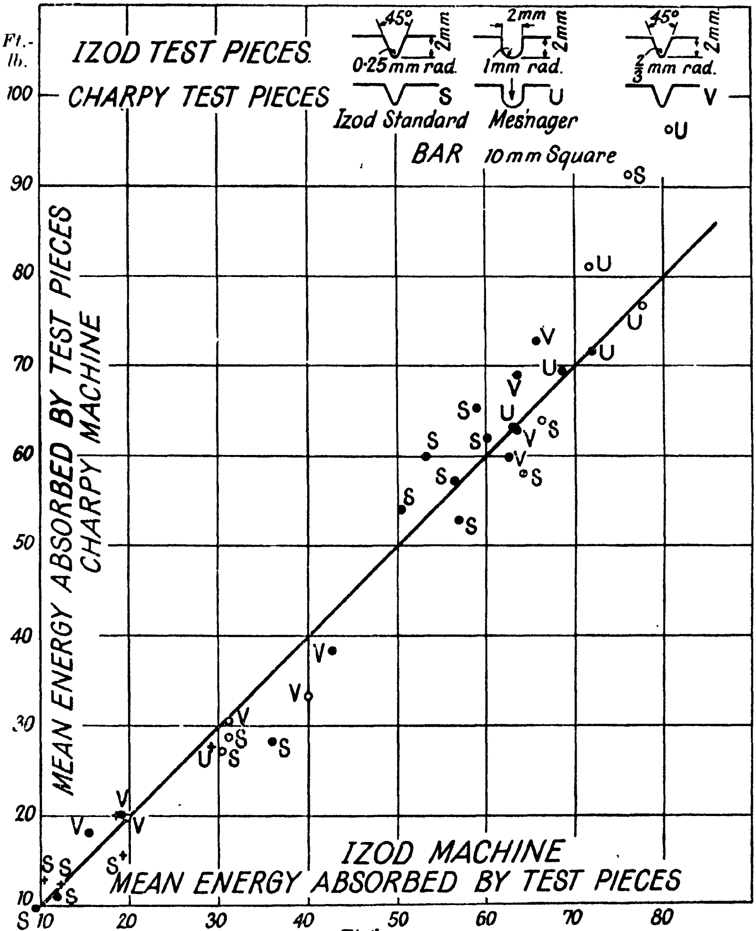


FIG. 140. COMPARISON BETWEEN IZOD AND CHARPY TESTS
(*Philpot*)

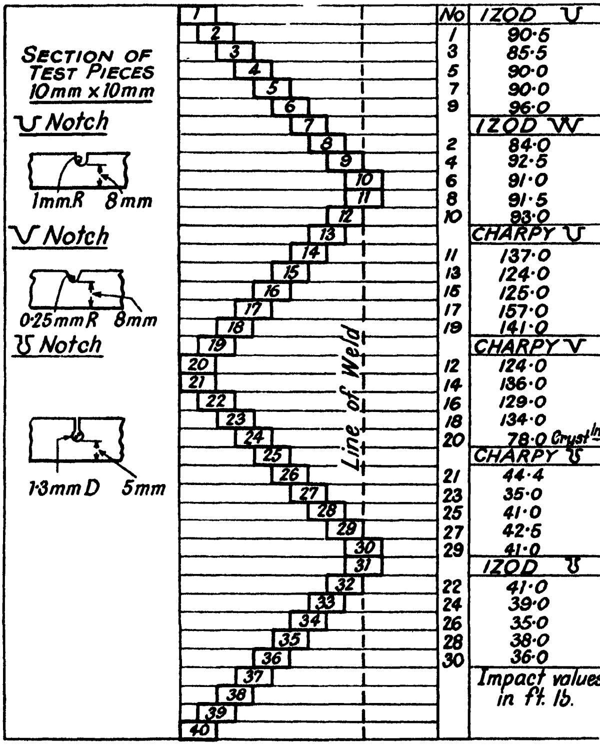


FIG. 141. RESULTS OF TESTS MADE IN IZOD AND CHARPY MACHINES

Tests in an Izod machine using steels of the type—

Percentage Composition							Yield Point Tons per in.²	Max. Load Tons per in.²
C	Si	Mn	S	P	Ni	Cr		
0·35	0·15	0·4	0·03	0·03	3·5	0·8	over 40	55 to 60

give Izod (Mésnager milled notch) = Izod (Standard) + 14·3.

Over 55 ft.-lb. the constant is 8·5.

The above relations are deduced from tests by Greaves and Moore.

For aircraft steels, tests by Philpot show that with the Izod machine.

Charpy (Mésnager) = Standard + 12 ft.-lb.

Charpy standard ($\frac{2}{3}$ mm. radius notch) = Standard + 7 ft.-lb.

With the standard 45° V-notch, 0·25 mm. root radius, heat-treated alloy steels give the same values in both machines up to about 50 ft.-lb.; above this the Charpy values exceed the Izod. Mild steels give about equal values in the two machines in the lower region but above this values are less in the Charpy

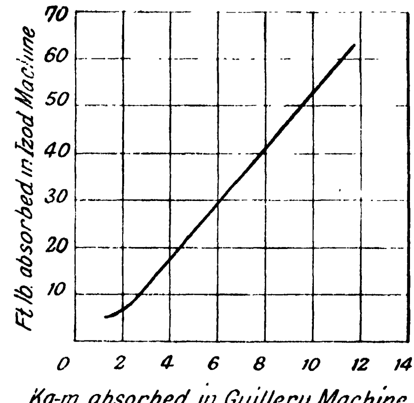


FIG. 142. CONVERSION CURVE
Guillery and Izod tests.
(*Greaves*)

than in the Izod. The values for medium carbon steels are generally less in the Charpy than in the Izod. Impact values for some commercial steels are given in Table XXII.

The Nature of the Notched-bar Test. The real meaning of the notched-bar test is as yet little understood. That the test is of value in providing an indication of incorrect heat treatment, or of disclosing a condition of embrittlement, or in ascertaining the ability of a material to withstand shock and stress concentration, is generally agreed. From a practical standpoint, if a test is found to be of use in distinguishing bad material from good or enables undesirable qualities to be eliminated, it is all that matters. This, however, is insufficient

to satisfy the scientific investigator and much work has been carried out with a view to discovering the principles underlying the test. The small measure of success which has attended these efforts is somewhat disconcerting: This may be in part explained by the fact that the complexity of the stresses set up in the specimen precludes the formulation of a useful mathematical theory, although attempts in this direction have been made. The test, moreover, is susceptible to the influence of a

(*a*)

(*b*)

FIG. 143. FRACTURES OF (*a*) TOUGH AND (*b*) BRITTLE NOTCHED BARS

number of variables each leading to a range in which test values are uncertain.

The impact figure, it is found, does not appear to depend on any dimensional law, comparable results not being given by specimens of different sizes. This is usually explained by the statement that similarity of dimensions does not extend to the grain size. Tough and brittle notched bars after testing have the appearance shown in Fig. 143. A tough specimen bends considerably before fracture and some energy is expended in bending portions of the test piece away from the fractured section and in dragging them between the supports. The two types of fracture are sometimes referred to as deformation fractures and separation fractures; the first for the tough and the second for the brittle material. Now, although this is taken as indicative of a certain difference of condition between the two materials, the same state can be frequently simulated in one and the same material by suitably varying the sharpness of the notch, or, again, by varying the width of the specimen, and every graduation between the extreme separation and deformation types of fracture may be exhibited.

The velocity of the hammer also affects the test and this combined with other factors sometimes introduces undesirable uncertainty. While it is known that a big change in the velocity had considerable influence on the result of an impact test, over a limited range the effect is often negligible. Comparatively little attention was paid to this factor, which tended to be obscured by the use of impact machines operating at fixed velocity and by the fact that almost invariably, as was

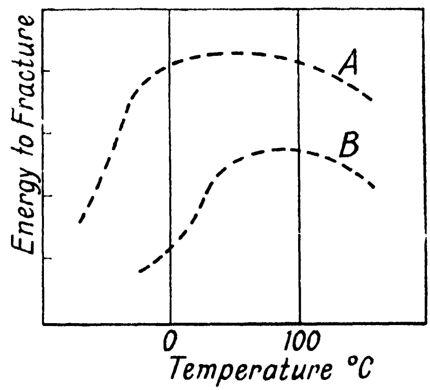


FIG. 144. SHOWING TEMPERATURE EFFECT

natural, the attention of workers in this field was concentrated on the properties of the material being tested. Nevertheless, it was known to some and suspected by others that, even in the range of velocity normally adopted, results were obtained which suggested the existence of a velocity effect. Later work has thoroughly established this.

It was also found that with some materials the energy absorbed by notched-bar test pieces varied with the temperature even within the range of the temperature of the room. The curves in Fig. 144 are illustrative. For material such as is represented by curve A, the impact value is practically constant throughout the whole range of temperature to which a room is normally subjected. Material of the kind indicated by curve B, on the other hand, is sharply critical to room temperature. This is at a fixed velocity.

In cases where the velocity of impact can be varied over a sufficient range a point is reached at which the energy of fracture shows a sudden fall in the manner indicated in Fig. 145.

The point of transition varies according to the kind and condition of the steel, and a particular heat treatment may cause the transition point to fall within the region of the velocity adopted in a standard machine.

Hence if two materials are being compared for impact one may show up much worse than the other, whereas, tested at a different velocity or at a different temperature, their respective impact values would show less difference. Since it is manifestly impossible to chart the effects of these variables to cover the

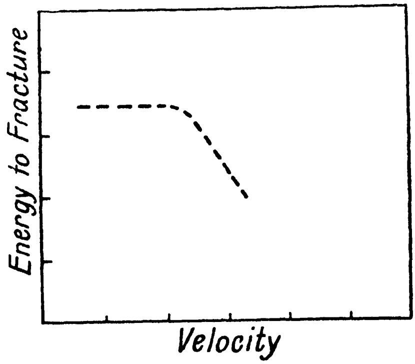


FIG. 145. SHOWING VELOCITY EFFECT

existing body of steels in all shades of graduation through heat treatment, in cases of suspicion, experience and judgment must be the guide.

At the other end of the velocity scale, if the notched bar be bent as a beam with the notch on the tension side and the test be performed slowly, when, of course, it ceases to be an impact test, a graph of corresponding loads and deflections may give more useful information than the impact numeral.

The work done in breaking a notched-bar test piece may be regarded as made up of (1) the work necessary to produce the maximum elastic deflection, (2) that necessary to deform the material until a crack begins, which is plastic strain, and (3) the work necessary to cause the crack to spread.

The impact test gives only the total energy whereas the slow-bend test gives some indication of the point where the crack commenced. The area under the curve when sufficiently definite represents the energy absorbed up to fracture as in the tension test (Fig. 146).

The result of a slow-bend test, while not strictly comparable with that of an impact test in a pendulum machine, nevertheless, in many cases, may show a good deal of correspondence. In contrast, materials such as overheated mild steel are tough in the slow-bend test and brittle under impact.

The subject of notched-bar testing was thoroughly discussed at a meeting in 1937 of a Joint Committee on Materials and their Testing held under the auspices of the Manchester Association of Engineers. The chief points arising out of this

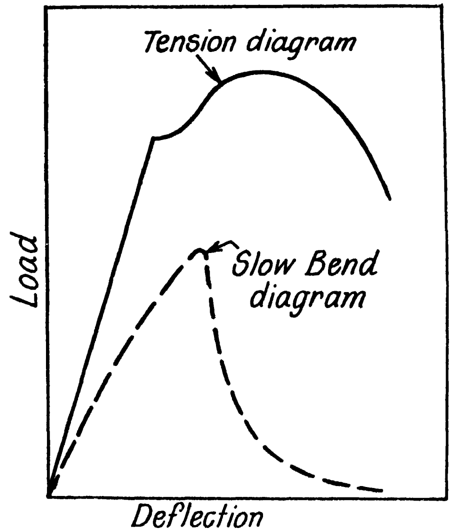


FIG. 146. SLOW-BEND AND TENSION DIAGRAMS

discussion are clearly set forth in a useful little book on *Notched-bar Testing*, by Mr. Leslie H. Hounsfield, M.I.Mech.E., which contains in addition an analysis of the Slow-bend Test.

The practical aspects of testing shed no light on the fundamentals of the notched-bar test, and the absolute meaning of the impact figure is not understood. Recently, Professor Southwell and his collaborators at Oxford have attacked the problem anew.

At the outset objection was taken both to the pendulum machines commonly used and their test pieces. The rigid

construction of the machines permits a transmission of energy by stress propagation to the earth, energy which is lost to the impacting system. Since this energy cannot be measured it has to be included in the impact value, and accordingly the value so obtained may be expected to be an overestimate. Further, in the Izod test piece the resultant action at the notched section has a shear component superposed on the bending moment, and in the Charpy specimen the striker may damage the specimen at the point opposite the notch and "the stresses brought into existence can only be conjectured."

In order to eliminate the loss of energy through transmission to earth the Oxford investigators suspended the striker and anvil by wires, the apparatus being slung like a ballistic pendulum and arranged so that the force of impact acts at the centre of gravity in a direction perpendicular to that of the suspension.

To overcome the remaining objections they adopted a method of 4-point loading for the test piece. The hammer, instead of striking the test piece directly, impinges on a yoke supported on the test piece and making contact at two points, one on either side of the notch as in Fig. 147. The stress set up is then normal to the plane of the notch. The test piece is circular.

In a comparison with a standard Izod machine, modified so as to bring the centre of percussion of the pendulum into coincidence with the point of impact, the values given by this modified Izod machine were in excess of those given by the Oxford machine to the extent of about 5 per cent.

It was also found that over the range through which experiments could be made, the energy of fracture was very nearly proportional to the area of the fractured specimen. The explanation offered is on the following lines.

The stress across the plane of the notch being wholly normal, fracture occurs only when the cohesion of the material is overcome. While the start of the fracture may be conditioned by the type of notch, the width of the section is negligible at the commencement, and from this point onwards the stress concentration is determined only by the sharpness of the natural crack. The natural crack, being very sharp, causes a high intensity of stress at the root so that the shape of the section can have little effect.

Plastic slipping in the crystals adjoining the root of the crack must be extremely small, and thus all occurrences are

confined to a thin layer containing the surface of fracture. In consequence the work done to fracture may be reasonably expected to conform to an "area" law.

Whether it is more rational to express the energy of fracture per unit volume or as the energy per unit area of cross section is a question which in the past has been productive of some controversy, opinion favouring an "area" rather than a "volume" law. Apart from the work described above, the law

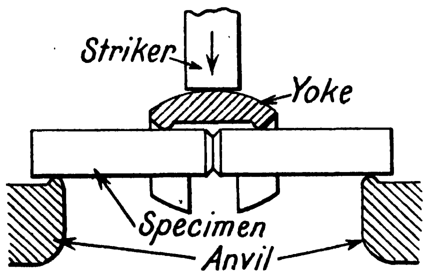


FIG. 147. OXFORD TEST PIECE: METHOD OF 4-POINT LOADING

which seems best to fit the case takes the form $E = kA^n$, where E is the energy to fracture, A the area of cross section, and k and n constants depending on the material, type of test piece, and notch.

Southwell's own view of the purpose of an impact test is that it is to measure the resistance of a material to the spread of a natural crack, that is, its power to resist fracture as contrasted with plastic distortion. He suggests that there is a quality of materials which is distinct from ductility, and is of importance in parts which are liable to shock because it controls the spread of a natural crack.

Authorities differ here, some holding that it is the property which determines the initiation of the crack which is preeminent. However this may be, the underlying principles have yet to be elucidated.

The influence of the factors referred to above is discussed by Lee in a paper, "The Notched-bar Test," in the *Journal of the Institution of Mechanical Engineers*, Vol. 143, p. 114.

The effects of temperature and velocity, in so far as they

concern practical testing, are concisely summarized in the following extract from *The Metals Handbook**—

"The effect of variation of temperature in notched-bar testing is extraordinarily great. The laboratory temperature may vary as much as 30° between summer and winter, and with some steels this is sufficient to affect the impact values materially. To obtain reproducible results in some cases requires the holding of the temperature of the test bar to a range which does not exceed 10° F."

And again—

"Inasmuch as most notched-bar testing is done on impact machines the velocities of which do not vary materially, the effect of velocity on reproducibility is often significant. Considerable work has shown that with most steels about the same energy absorption is secured whether the bar is broken statically, or by impact at normal speed. On the other hand, it has also been shown that some steels are velocity sensitive, and when broken statically they remain tough in wider sections or with sharper notches and at lower temperatures than when broken by impact. This field of sensitivity requires more study than has been put on it before an adequate discussion can be given."

* 1939 edition. Published by The American Society for Metals.

CHAPTER X

REPEATED STRESSES

Fatigue of Metals. The extensive series of tests carried out by Wöhler between the years 1860–1870 on the effect of repeated stresses on materials showed clearly that a completely reversed stress much lower than the ultimate strength of the material, or lower even than the yield point as determined by the tensile test, could cause fracture of a steel specimen if only the application of stress was repeated a sufficient number of times. On the other hand, below a certain maximum value of the stress Wöhler showed that the number of repetitions of stress might be indefinitely large without causing rupture.

The phenomenon of fracture under repeated stressing is termed *fatigue*.

Formerly, it was held that under repeated stress a metal developed a crystalline structure, but later metallographic research has shown that metals are themselves built up of crystalline grains and that, therefore, crystallization of the material is not a consequence of repeated stressing. (See Chapter 11.)

The impression that crystallization was produced in a metal by the action of repeated stresses arose largely from the brittle appearance of such fractures, and from the fact that many of these exhibited a coarse crystalline structure. This appearance may have been due to the heat treatment received by the material during the process of manufacture.

A mild steel specimen when tested in tension suffers considerable plastic flow prior to fracture and the surface of the ruptured section shows a silky, fibrous structure, owing to the crystals having stretched in the direction of the pull. A fatigue crack, however, has an entirely different appearance. Owing to a local defect, or to the action of fatigue in causing hair cracks to form in the material, the concentration of stress at the end of the crack under the stressing action causes the crack to spread progressively until the cross section becomes so reduced that the remaining portion fractures suddenly under the load imposed.

In general, fatigue failures of ductile materials show two distinct zones; one exhibiting a characteristic brittle appearance

—the fatigue fracture proper; the other exhibiting a more or less ductile fracture somewhat similar to that shown under a tension test. The difference between the two zones is not, as a rule, so apparent in the more brittle materials. Fatigue

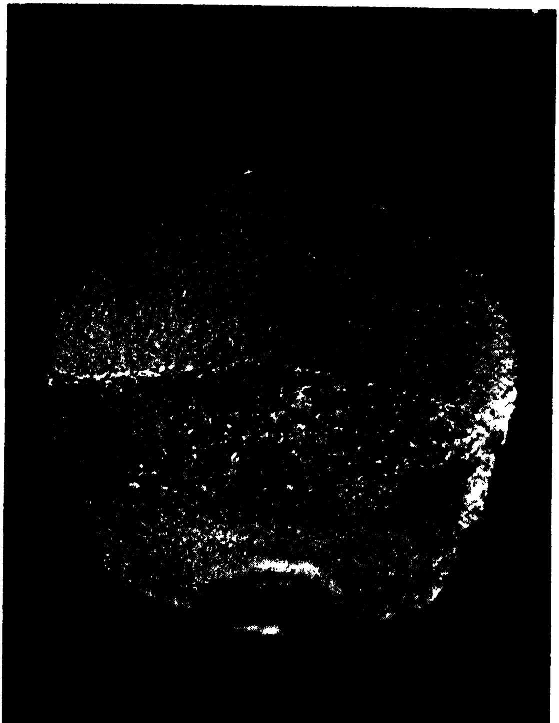


FIG. 148. FATIGUE FRACTURE OF MILD STEEL TEST PIECE
(Bruntons (Musselburgh) Ltd.)

fractures are sometimes discoloured by chemical action. Fig. 148 shows a fatigue fracture of a mild steel test piece.

The shape of the zones appears to depend on circumstances attendant on the material and the mode of stressing. Professor Bacon has classified the fatigue fractures met in practice as follows (see Fig. 149)—

(*a*) Concentric.
(*b*) Eccentric.
(*c*) Double sided, convex.
(*d*) Double sided, concave.
(*e*) Single sided, concave.
(*f*) Single sided, convex.

In the diagrams the white portions represent the fatigue zones and the black portions the ruptured core.

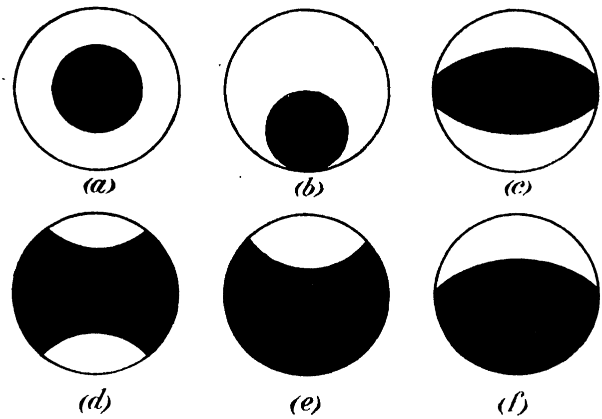


FIG. 149. ILLUSTRATING TYPES OF FATIGUE FRACTURE

In some cases many fatigue zones are apparent. The arcs separating the respective zones are generally elliptical in shape. The diagram in Fig. 150 represents a Hypothetical Crack Progress Chart, which consists of a system of elliptic arcs starting from a small semicircle centred at the origin of the crack at A and eventually straightening out into a diameter tilted back through 15° to the direction of rotation. Later work indicates that high stresses produce concentric fractures while low stresses produce eccentric fractures.

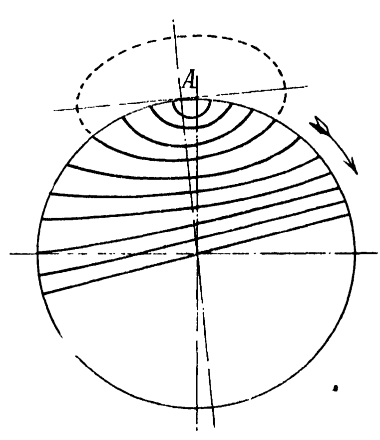


FIG. 150. HYPOTHETICAL CHART SHOWING PROGRESS OF A FATIGUE CRACK

In rotating bar test pieces the maximum stress occurs at the outer fibres, and the fatigue crack usually starts at the periphery and spreads towards the centre. When there is concentration of stress due to fillets or holes, the crack

usually starts at the most highly stressed portion and spreads from this point.

Work of Bauschinger and Bairstow. One of the first investigators of cycles of stress was Bauschinger, a contemporary of Wöhler. He loaded and unloaded specimens slowly and determined the stress-strain relation under these conditions by using a very sensitive extensometer. He showed from his tests that the proportional limits in tension and compression are not fixed points for a given material and that they may be displaced by submitting a specimen to cycles of stress.

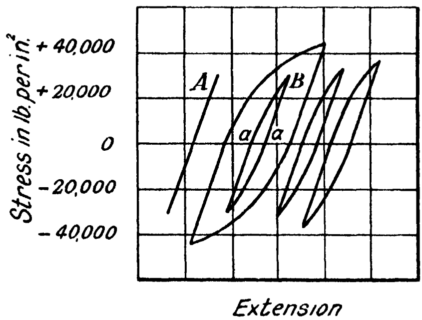


FIG. 151. HYSTERESIS LOOPS OBTAINED UNDER CYCLICAL STRESS

To explain why the endurance limit for steel under reversed stress was lower than the limit of proportionality obtained in the static test, Bauschinger advanced the theory that the material as received from the manufacturer might have its proportional limits in tension and compression raised by cold work, and that the true or natural proportional limits are those which are established after submitting the material to cycles of stress. These natural proportional limits are supposed to define the safe range of stress in fatigue tests.

The work of Bauschinger was extended by Bairstow, who used a slow loading and unloading machine (2 cycles per minute) with a mirror extensometer, and obtained the stress-strain relations for cycles with various ranges of stress. Some of Bairstow's results are shown in Fig. 151. The material was axle steel possessing a yield point of 50 000 lb. per in.² and an ultimate strength of 84 000 lb. per in.² The cycle consisted of equal reversed stresses. The line *A* represents the initial tension—compression test with the range 31 400 lb. per in.², within these limits the straight line relationship holds.

The specimen was next subjected to cycles of reversed stress of 31 400 lb. per in.² when it was found that the straight line developed into a loop. Curve *B* shows the loop obtained after 18 750 cycles. It will be noted that in this case the initial

proportional limits are higher than the so-called "natural" limits obtained after many cycles of reversed stress, and that a cyclical permanent set of width *aa* was produced. The remaining loops were obtained after a number of cycles of reversed stress equal to 33 500, 37 500 and 47 000 lb. per in.^{2} respectively, sufficient apparently to stabilize the size of the loops.

When the width of these loops was plotted against the corresponding maximum stress Bairstow found that the results of these experiments gave approximately a straight line. The intersection of this line with the stress axis determines the range of stress at which there is no looping effect.

The range of stress so defined was assumed to be the safe range of stress and subsequent endurance tests have tended to verify this assumption.

Mechanism of Fracture. The first attempt to explain the mechanism of fracture in endurance tests was made by Ewing and Humfrey. They used a rotating specimen of Swedish iron with a polished surface and examined this surface microscopically after applying cycles of reversed stress. They found that if stresses above a certain limit were applied, slip bands appeared on the surface of some of the crystals after a number of cycles. The number of slip bands increased as the cycles were repeated and some of the previous slip bands seemed to broaden out. This broadening process continued until a crack occurred, the crack following the marking of the broadened slip bands. They found that a reversed stress of 11 800 lb. per in.^{2} could be applied millions of times without producing any slip bands. On the basis of their investigations they advanced the theory that cycles of stress which are above the safe range produce slip bands in the individual crystals. If the application of stress cycles is contined, sliding along the surfaces takes place accompanied by friction. According to the theory, as a result of this friction the material gradually wears along the surface of sliding and a crack results. Further investigation showed that slip bands occurred at stresses much lower than the endurance limit of the material. They may develop and broaden without leading to the formation of a crack. The appearance of slip bands cannot therefore be taken as a criterion for determining the endurance limit.

A vast amount of data concerning strength in fatigue has been accumulated in recent years, but up to the present no theory has been established which is wholly satisfactory to

engineers in its explanation of the cause and mechanism of fatigue failure.

Stress Cycles. Fatigue Limit. The type of stressing producing fatigue fractures may be tension and compression, bending, shear, torsion or a combination of these. The cycle of stress need not consist of reversed tension and compression stresses of equal magnitudes. For example, the cycle may vary from a stress of 5 tons per in.^{2} in tension to 10 tons per in.^{2} in compression, or the stress may be wholly tensile but of variable magnitude; or, again, the stress may vary in torsion from a given value to a different value in the opposite direction.

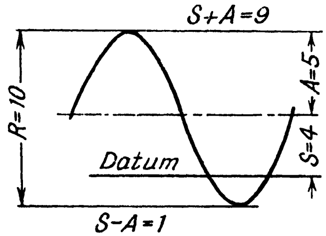


FIG. 152. SIMPLE HARMONIC STRESS CYCLE

Frequently, both in practice and under test conditions, the magnitude of the stress when plotted on a time base gives a sine curve. Even if this is not the case the stress variation occurs in cycles and it is customary to refer to the repeated variation as a *stress-cycle.*

The stress throughout the cycle may be regarded as made up of a mean steady stress S and an alternating stress A. The diagram, Fig. 152, illustrates a simple harmonic stress cycle composed of a steady stress S = 4 tons per in.^{2} combined with an alternating stress A = 5 tons per in.^{2}, giving 9 tons per in.^{2} in tension and 1 ton per in.^{2} in compression. If R be the algebraical difference between the maximum and minimum values of the stress then R is termed the *range of stress.* The limiting values of the stress are thus S + A and S − A.

The range of stress within which an indefinitely large number of repetitions of stress will not cause fracture is called the *fatigue range.* The *fatigue limit* is the greatest stress that can be applied in a given stress cycle without eventually causing fracture.

The *endurance* is the number of cycles required to cause fracture when the range of stress is maintained invariable throughout the test. The fatigue limit is by some authorities referred to as the *endurance limit.*

The value of the fatigue limit appears to be independent of the speed or frequency, at least up to a few thousands of cycles per minute. Excessively high frequencies may tend to cause

heating of the specimen through hysteresis. Some results of extremely high speed tests will be given later.

The Wöhler Test. Effects of Fillets and Surface Finish. Obviously there are many ways of applying a varying stress cycle to a test piece with more or less satisfactory experimental results, but from the point of view of commercial testing the two outstanding methods are—

(1) The Wöhler test in which a load is applied to a rotating bar, stressing it in bending, the outer fibres suffering alternate tension and compression as the bar rotates.

(2) Subjecting the specimen to alternate tension and compression as in the Haigh machine.

One objection urged against the Wöhler test is that it is merely a skin test, since the major part of the section is but comparatively lightly stressed. This type of stressing is, however, that sustained by innumerable machine components in practice and is of value on that count alone. Moreover, the apparatus needed to make the test is comparatively cheap and easy to set up. Probably the bulk of fatigue testing has been done by this method.

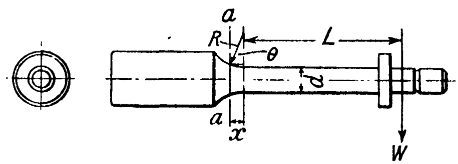


FIG. 153. WOHLER TEST PIECE

The Wöhler test piece is shown in Fig. 153. The shank is gripped in a chuck provided with adjusting screws while the free end carries a ball bearing from which the load is suspended. The inner race of the ball bearing may be a push fit on the specimen, but to avoid possible trouble during running it is advisable to thread the end of the test piece and to clamp the inner race with a suitable sleeve and nut.

To avoid a sharp discontinuity, the body of the specimen is joined to the shank by a fillet. The radius of the fillet has considerable influence on the results obtained.

One effect of the radius is to cause the maximum stress to occur at a section other than at the junction. If d is the diameter of the parallel portion of the specimen, r the radius of the fillet and L the distance from the line of action of the load to the junction, then the distance x from the junction of the section at which the greatest stress occurs is very approximately

$$x = rd/6\mathrm{L}$$

For example, if L = 2 in., $d = \frac{5}{16}$ in.; $r = \frac{5}{8}$ in., then $x =$ 0·0163 in.—less than 1 per cent of the nominal length. If L = 4 in.; $d = \frac{1}{2}$ in.; $r = \frac{3}{8}$ in.: then $x =$ 0·0078 in., which is less than 0·2 per cent of the nominal length.

The smaller the radius of the fillet the more nearly will the section at which the stress is a maximum coincide with the section of the junction, but this is attended with some disadvantage owing to stress concentration causing the value of the fatigue limit to be different from that calculated from the ordinary formula for bending. For comparative purposes it may be of less moment; in fact, one investigator at least has used a fillet radius of one-tenth diameter of the test piece.

The effect of the radius of the fillet has been investigated experimentally, and some results for carbon steel are given in the following tables—

TABLE XI

EFFECT OF RADIUS OF FILLET ON FATIGUE PROPERTIES OF 0·49 PER CENT CARBON STEEL SPECIMENS

(Moore)

Radius of Fillet r in.	Minimum Diameter of Specimen d in.	Ratio r/d	Fatigue Limit lb. per in.^{2}	Reduction in Fatigue Limit %
9·85	0·275	36	49 000	—
1·00	0·275	3·5	47 500	3
0·275	0·275	1	44 500	9
0·00	0·275	0	24 000	51

TABLE XII

EFFECT OF RADIUS OF FILLET ON FATIGUE PROPERTIES OF 0·33 PER CENT CARBON STEEL SPECIMENS

(Timoshenko and Lessels)

Radius of Fillet r in.	Minimum Diameter of Specimen d in.	Ratio r/d	Fatigue Limit lb. per in.^{2}	Reduction in Fatigue Limit %
Standard Form	—	—	32 000	—
0·15	0·6 parallel	1/4	29 900	6·5
0·05	0·6 parallel	1/12	21 000	34

These results show the influence of change of section on the value of the fatigue limit as experimentally determined, particularly when the ratio r/d falls below unity. The point is important in relation to any proposed standardization of the Wöhler test piece.

A method of securing a more uniform distribution of stress along the test piece is to employ the tapered form of Fig. 154.

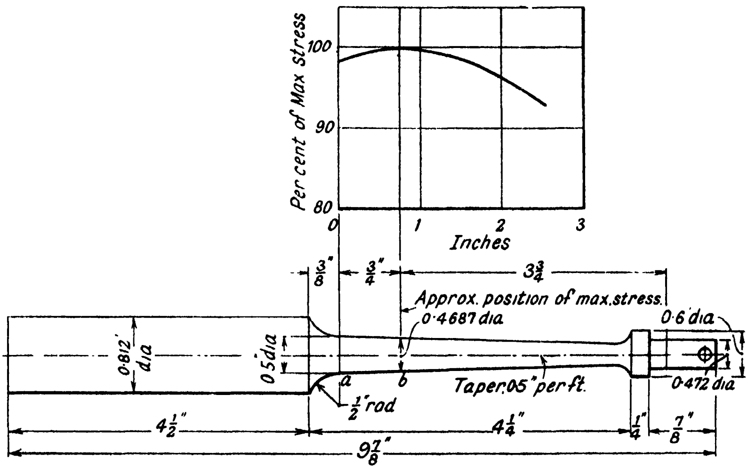


FIG. 154. WÖHLER TEST PIECE.
Tapered form.

The taper is 0·5 in. per ft. and the variation of fibre stress over the length ab is only about 1 per cent. The variation of stress is shown in the graph in the same figure.

Specimens are sometimes made hollow at the shank end. The method is expensive as the bore needs to be finished smooth.

Another objection urged against the Wöhler test piece with single point loading (that is, with the test piece loaded as a cantilever) is that in addition to the bending stress there is also a direct shear on the section. This objection is overcome by the use of four-point loading in which the specimen is supported at the ends and loaded at two points equidistant from the supports, Fig. 155. This method of loading gives a constant bending moment over the gauge length with no direct shear on the section. A larger amount of material is required with this form of specimen unless the holders are designed to accommodate a short test piece. A modification of the foregoing is to

use the Wöhler test piece with two-point loading, Fig. 156. The downward load is applied by a dead-weight and an equal and opposite load is applied by means of a small single-lever testing machine. The bending moment $w \times l$ is constant over

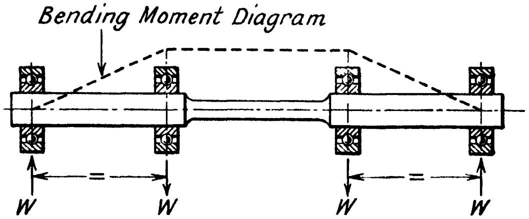


FIG. 155. SONDERICKER TEST PIECE
Four-point loading.

the test portion. In any case the amount of direct shear is very small. A $\frac{5}{16}$ in. diameter specimen 2 in. long carrying a single load of 40 lb. at its free end is stressed in bending to about 12 tons per in.^{2}, whilst the mean shear stress over the section is only 500 lb. per in.^{2}, which is less than 2 per cent of the bending stress.

The degree of finish given to specimens is also of importance. Kommers found that test pieces of ordinary mild steel finished

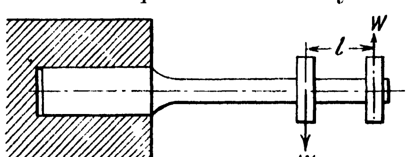


FIG. 156. WÖHLER TEST PIECE
Two-point loading.

by hand filing, had a 33 per cent longer life than when merely turned, and that this increase amounts to about 48 per cent when specimens are polished.

The oft-quoted results of Professor W. Norman Thomas are further confirmation of this. Gelatine casts were taken from a number of specimens of various degrees of surface finish. The casts were sliced with a microtome and the depth of scratch and radius of curvature at the root measured by the aid of a projection apparatus. The results of fatigue tests on scratched and highly polished specimens were compared, and comparison was also made with deductions from mathematical theory.

TABLE XIII

EFFECT OF SURFACE FINISH ON THE ENDURANCE OF MILD STEEL

Type of Finish	Estimated Reduction in Endurance Limit %
Turned . . .	12
Coarse file . . .	18–20
Bastard file . .	14
Smooth file . .	$7\frac{1}{2}$
No. 3 emery . .	6
No. 1 emery . .	4
No. 0 emery . .	2–3
Fine carborundum .	2–3
Fine ground . .	4
Accidental scratches (maximum found) .	16

Endurance Curves. A typical endurance curve for carbon steel is given in Fig. 157. The curve tends to show that at a sufficiently low stress the material would withstand an in-

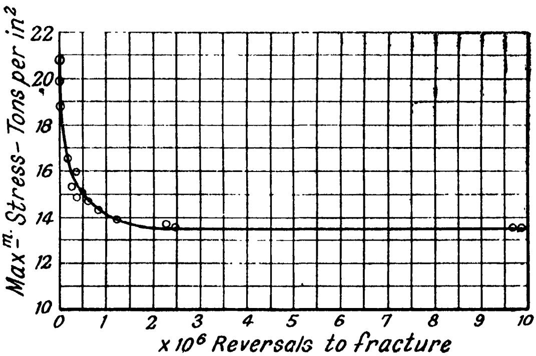


FIG. 157. ENDURANCE TEST ON 0·83 PER CENT CARBON STEEL

(*Moore*)

definite number of reversals. For some steels, however, and for non-ferrous materials the endurance curves approach the axis of abscissae very slowly, indicating that any varying load, however small, would ultimately cause fracture of the material. Fatigue tests in which there is a combination of the effects of stress and corrosion lead to a similar result. It should be remarked, in passing, that failure of a metal under repeated

stress is rapidly accelerated when, in addition, corrosive influences are at work. (See Fig. 194.)

Alloy steels, especially those containing a high nickel content, often give no "curve," but show an abrupt change in the region of the fatigue limit.

Fig. 158 refers to some results obtained by Moore on a 0·83 per cent carbon steel, some specimens of which remained unbroken after 100 million reversals of stress. It will be observed that in the neighbourhood of the fatigue limit a small change in

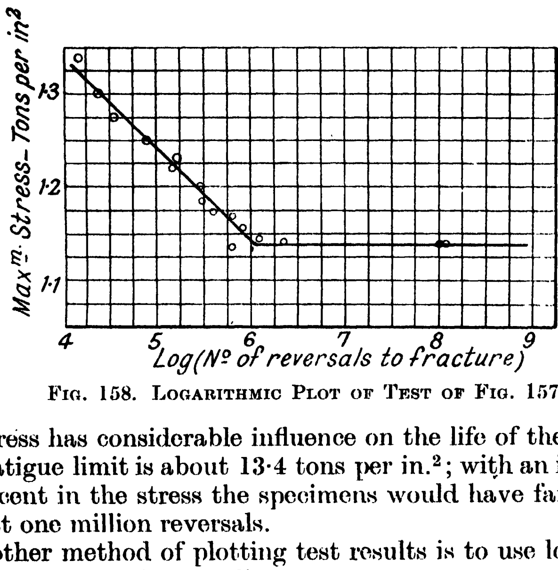


FIG. 158. LOGARITHMIC PLOT OF TEST OF FIG. 157

the stress has considerable influence on the life of the material. The fatigue limit is about 13·4 tons per in.^{2}; with an increase of 3 per cent in the stress the specimens would have failed, probably at one million reversals.

Another method of plotting test results is to use logarithmic or semi-logarithmic co-ordinates.

Fig. 158 is plotted from the results already given in Fig. 157; the logarithms of the stresses being plotted against the logarithms of the numbers of reversals. The abrupt change at the fatigue limit is particularly noticeable.

The ratio of the fatigue limit to the ultimate strength in the case of steels varies from about 0·35 to about 0·65.

Wöhler Fatigue Testing Machine. The general arrangement of a Wöhler machine employing single-point loading is shown in Fig. 159. Two specimens can be tested at once and the machine can be arranged to accommodate specimens of 2 in. and 4 in. or other convenient test lengths. The drive is by electric motor at 2 000 r.p.m. A Veeder counter is provided for each spindle

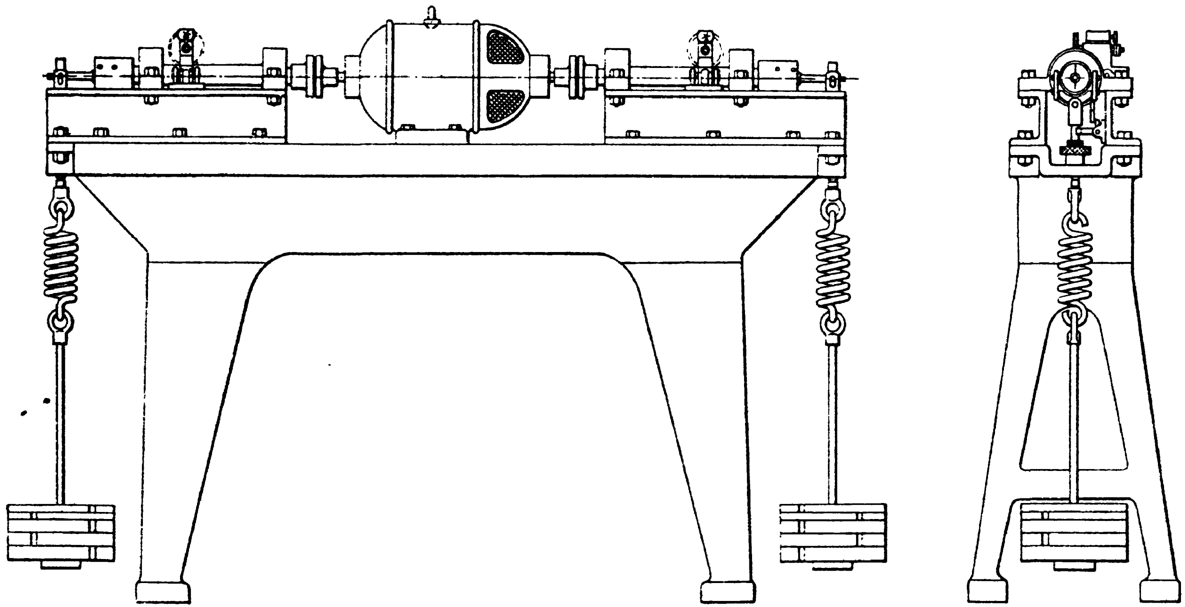


FIG 159. WÖHLER TESTING MACHINE
(Machinery)

and is driven through a 100 to 1 reduction gear. The load is applied by slacking back the hand nut seen in the end view.

A spring or a strip of rubber may be inserted between the deadweight and the ballrace at the end of the specimen to assist in damping out any vibration caused by running, though this is negligible if the specimen is set up accurately. For this purpose an Ames dial may be used. Dashpots are sometimes

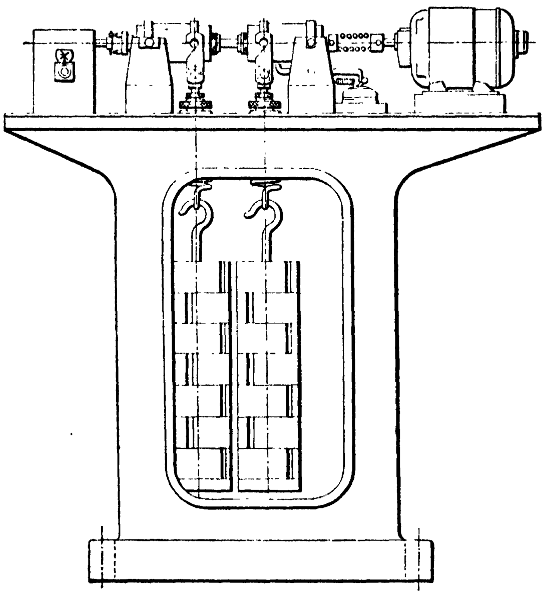


FIG. 160. MACHINE EMPLOYING FOUR-POINT LOADING

employed in place of the damping springs. The counter is tripped when the test piece fractures. Some experimenters view a double ended machine with disfavour and prefer to work with a single spindle.

Author's Fatigue Testing Machine. A machine employing four-point loading, designed by the Author, is shown in Fig. 160. The test specimen is carried between two spindles running in ball bearings mounted in housings pivoted on vertical supports, the drive being through a spring transmission from a high speed motor. The tail spindle communicates its motion to a substantial revolution counter. Both the motor and counter are cut out of action immediately fracture occurs.

A novel feature of the machine is the method of gripping the test piece. This is accomplished by means of special chucks which obviate the necessity for drilling and tapping the ends of the specimens; a distinct advantage when a large number of specimens have to be tested, particularly if they should be of hard or tough material.

Damping springs are provided between the loads and the supporting knife-edges. Provision is made for gradually applying and releasing the loads.

Specimens are 3 in. long and may vary from 0·2 to 0·3 in. diameter at their smallest section. Close limits in machining specimens to length are unnecessary as sufficient freedom of the housings is allowed to accommodate variations in the overall length of the specimens.

The type of specimen adopted and the method of gripping it enables alignment to be obtained with no trouble on the part of the operator. Specimens may be parallel in the mid-portion of their length or turned so that the longitudinal profile is a curve of some 3 in. radius as desired.

A load of 60 lb. with a specimen 0·3 in. diameter will yield a stress of 30·3 tons per in.^{2}, and with a specimen of 0·25 in. diameter will yield a stress of 51·3 tons per in.^{2}

The normal speed of operation is 3 000 r.p.m.

The machine is now made by Messrs. Edward G. Herbert, Ltd.

The Haigh Alternating Stress Testing Machine. The alternating stress testing machine designed by Professor Haigh is made by Messrs. Bruntons Ltd., Musselburgh. Two sizes are made, one having a load range of $1\frac{1}{2}$ tons and the other a load range of 6 tons. The important advantage of this machine lies in the uniform distribution of stress across the full section and along a considerable length of the test piece, and in the possibility of applying stress cycles of unequal plus and minus limits.

The essential features of the machine are shown in Fig. 161. A pair of electromagnets M_1 and M_2 are supplied with two-phase currents from a small alternator of special design. The forces generated by these magnets, pulling alternately on the faces of an armature A, are transmitted to the lower end of the test piece T, the upper end of which is gripped in a holder that forms part of the adjustable head H. The magnets and the adjustable head are rigidly connected by four vertical columns that rise from the base of the machine. The vibrating armature

is guided by springs without the use of lubricated slides. The frequency of reversal of stress is governed by the speed of the alternator, usually 1 000 r.p.m., and the frequency of stress reversals is double this, or 2 000 cycles per min.

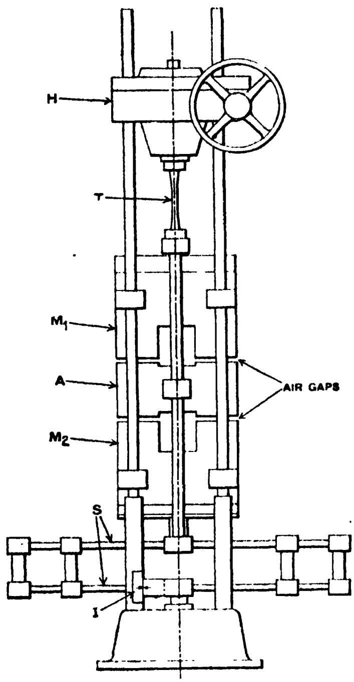


FIG. 161. PRINCIPLE OF THE HAIGH ALTERNATING STRESS TESTING MACHINE
(*Machinery*)

Each electrical cycle produces two mechanical cycles, i.e. pull—push—pull—push.

An important feature of the machine is the set of compensating springs *S* that connect the vibrating crosshead below the magnets to the base of the machine.

Although the majority of fatigue tests are performed with loads ranging between equal intensities of push and pull the ratio between the extremes may be varied at will. This adjustment is effected by giving the flat springs *S* a suitable degree of initial load. The same springs serve the important purpose of compensating the force required to accelerate the armature and other vibrating parts.

By means of sliding clamps the spring stiffness may be adjusted to suit widely different frequencies of operation.

The machine is stopped automatically when the test piece breaks.

The stress meter, Fig. 162, is a soft iron alternating current instrument comprising two independent movements in a single case. The outer, long, uniformly divided scale is used for reading the range of stress applied to the machine. The inner short scale, graduated to read amperes, is arranged with a central zero for the purpose of balancing the currents in the two phases to ensure that the air gaps are equal on the upper and lower faces of the vibrating armature.

The datum position of the compensating springs is that in which it supports the weight of the armature in its central position midway between the pole faces. To find the datum position a stiff test piece is inserted in the machine and a moderate stress, say three-quarters of full load, applied. The handwheel is adjusted to bring the ammeter needle to zero and the machine then stopped. The screws of the automatic cut-out are adjusted until they just touch the operating trigger in its central position.

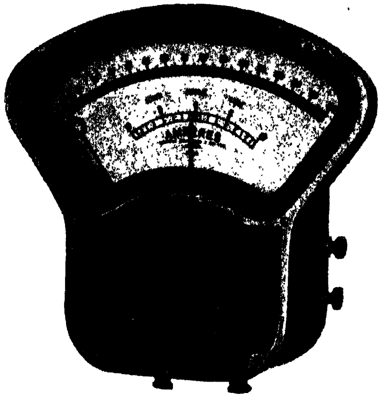


FIG. 162. STRESS METER EMPLOYED WITH THE HAIGH MACHINE
(*Bruntons (Musselburgh) Ltd.*)

The test piece is then removed and the trigger adjusted until it is in contact with both screws of the automatic cut out. The

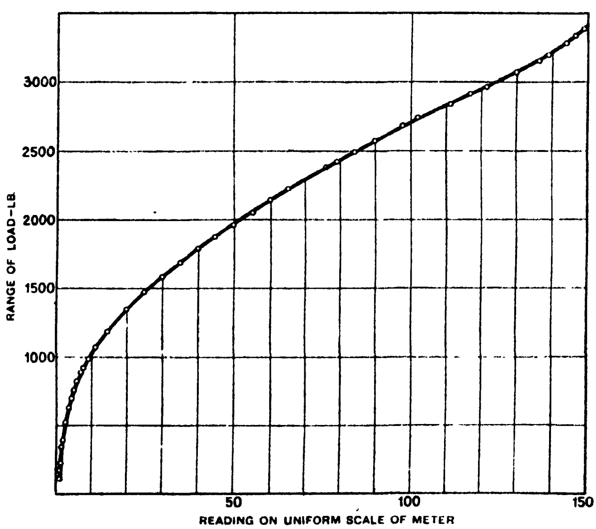


FIG. 163. CALIBRATION CHART FOR USE WITH HAIGH MACHINE
(*Machinery*)

screw is locked in this position and readings of the scale and micrometer head observed. This is the datum position. The stress on the specimen is obtained from a calibration chart similar to that in Fig. 163 in conjunction with the stress recorder.

The apparatus for calibrating the machine comprises a special extensometer fitted with a vibrating mirror and a camera.

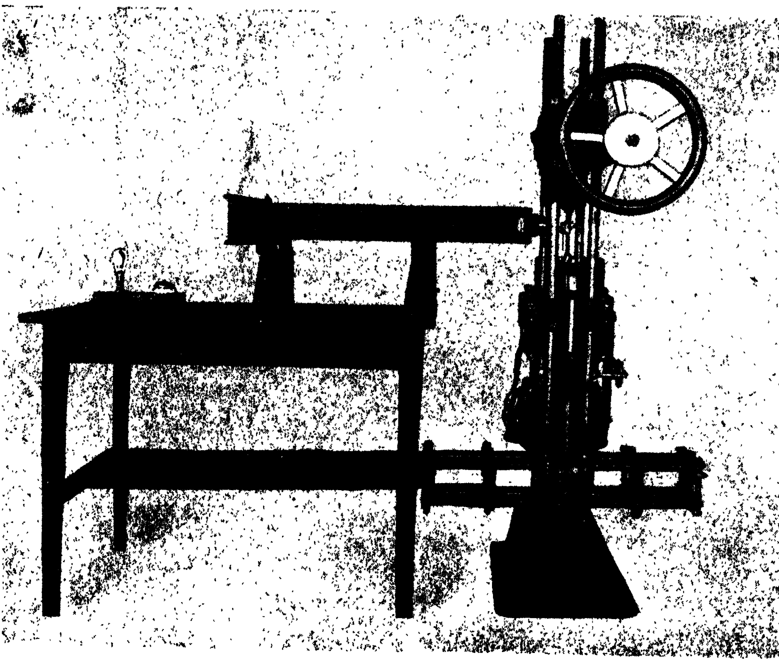


FIG. 164. MACHINE AND CAMERA SET UP FOR CALIBRATION
(*Bruntons (Musselburgh) Ltd.*)

The machine and camera set up for calibration are shown in Fig. 164.

The form of the test piece for standard investigations in the smaller model is shown in Fig. 165. Pieces up to 15 in. long can be tested if desired. The screwed ends may be an easy fit in the holders, but the threads should be true to ensure axial loading. The transition curve between the cylindrical mid-length and the tapered ends should be rounded off carefully.

With high grade uniform materials four or even three test

pieces may suffice to indicate the fatigue limit, but in less uniform materials a greater number of tests is desirable.

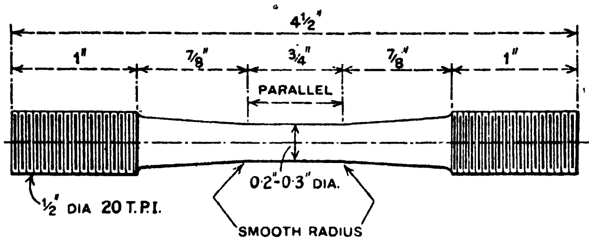


FIG. 165. FORM OF TEST PIECE
(*Machinery*)

Results of Tests with Haigh Machine. The results of some tests on high tensile brass are given in Table XIV and the corresponding graphs in Fig. 166. It will be noted that sample

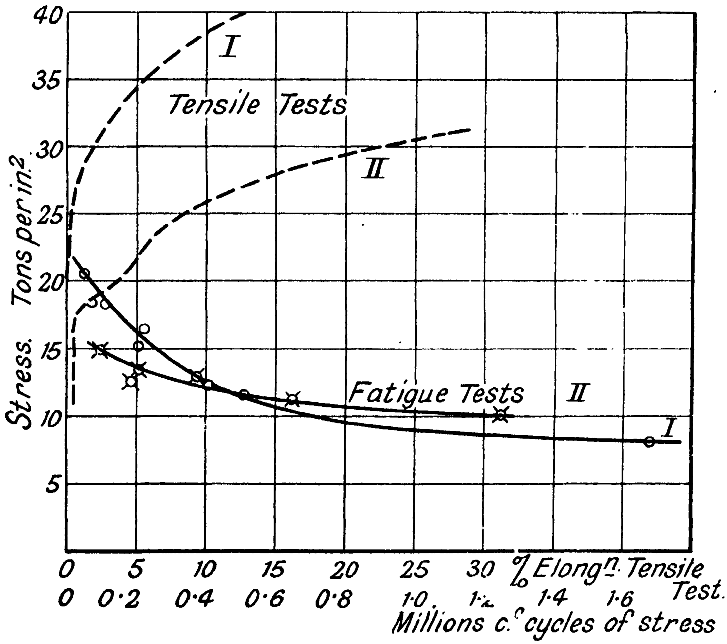


FIG. 166. RESULTS OF TESTS CARRIED OUT ON THE HAIGH MACHINE

TABLE XIV

Comparison of High Tensile Brasses (Haigh)

Sample	Tensile Test			Fatigue Test		
				2 000 cycles per min.		
				Limiting Stress (tons per in.^{2}) for Endurance of		
	Ultimate Strength Tons per in.^{2}	Yield Point Tons per in.^{2}	Elongation Percentage	100 000 cycles	390 000 cycles	1 500 000 cycles
I . .	40·5	27	15	18·5	12·5	8·2
II . .	31·5	18	29	14·7	12·5	10·0

No. 1 had an ultimate strength of over 40 tons per in^2. compared with 31 tons per in.^{2} for sample No. 2. The yieldpoint was also higher than in No. 1. So far as fatigue is concerned, No. 1 withstood heavy stresses for slightly longer than No. 2, but the minimum stress required to break No. 2 was actually higher than for No. 1. The fatigue limits were 8 and 10 tons per in.^{2} respectively.

In many tests only the fatigue limit for reversed stresses is determined. For purposes of design it is frequently desirable to know the limits of a material under varying stresses which are not completely reversed.

The safe limits of stress for unlimited endurance can be expressed as $S \pm A$ where $2A = R$ is the range of stress.

If f_a and f_i represent the maximum and minimum stresses

then
$$f_a = S + A$$
$$f_i = S - A$$
and
$$2A = f_a - f_i$$

In order that the ultimate strength, f, of a material may not be exceeded, S must not be greater than f and $S + A$ must not exceed f when $A \neq 0$.

Between the two limiting cases, namely, when the mean stress is equal to the ultimate tensile and compressive strengths, respectively, an infinite number of stress ranges exist in which the mean stress passes from a value in compression to a value in tension.

Goodman's Dynamic Law. The work of the early investigators showed that the safe range of stress for any particular value of the mean stress depended on the mean stress, and one law put forward was Goodman's Dynamic Law. It had its origin in the fact that a suddenly applied tension load produced an instantaneous stress twice as great as when the load was applied gradually. Under a varying stress, f_a and f_i are the maximum and minimum stresses in any safe range. The range

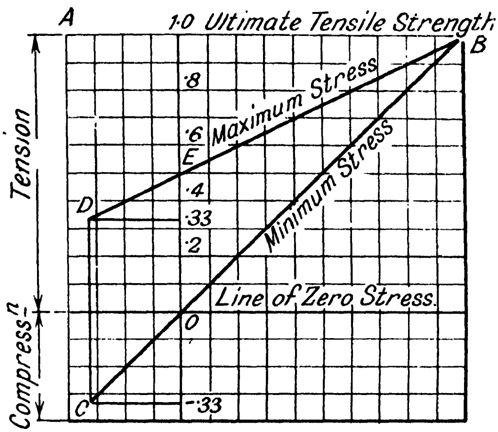


FIG. 167. DIAGRAM ILLUSTRATING GOODMAN'S DYNAMIC LAW

of stress $(f_a - f_i)$ is regarded as a load suddenly superposed on f_i, the minimum value. For the ultimate strength not to be exceeded

$$f_i + 2(f_a - f_i) \not> f$$

or

$$2f_a - f_i \not> f$$

Tension and compression are here indicated by + and − signs respectively.

In the diagram, Fig. 167, the line AB represents the ultimate static strength. Minimum stresses are plotted along BOC. According to the theory, the maximum applied stress should fall on a line DEB such that the point E is 0·5 of the ultimate static strength.

For equal reversed loads it will be noted that the safe range of stress is from + 0·33 to − 0·33 of the ultimate strength.

This law is not of general application.

Gerber's Parabolic Law. It was suggested by Gerber that all safe ranges of stress are related to the corresponding mean stress and to the ultimate strength by the relation

$$f_a = A \pm \sqrt{(f^2 - 2nAf)}$$

where n is an experimental constant.

The foregoing relation can be transformed thus —

$$(f_a - A)^2 = f^2 - 2nAf$$

that is,

$$S^2 = f^2 - nRf$$

S being the mean stress for any safe range R.

If we put $S = 0$ we obtain

$$n = \frac{f}{R'} = \frac{\text{ultimate strength}}{\text{safe range for reversed stresses}}$$

Consider again the relation

$$S^2 = f^2 - nRf$$

We have $\quad nRf = f^2 - S^2,$

therefore

$$\begin{aligned} R &= f^2/nf - S^2/f \\ &= (f/n)(1 - S^2/f^2) \\ &= R'(1 - S^2/f^2) \end{aligned}$$

where $R' = f/n$ is the safe range for reversed stresses.

Modified Goodman Law. A modified Goodman law has been introduced in which the ratio of the safe range of stress for reversed stresses to the ultimate strength of the material can have any value, but other ratios diminish regularly as the ultimate strength is approached.

This law can be expressed by

$$R = R'(1 - S/f)$$

the symbols having the same meanings as before.

A diagram can be plotted, Fig. 168, to show the safe range of stress for various values of the mean stress of the cycle. The diagram is seen to consist of a number of isosceles triangles whose apices lie on the vertical through $S = 0$. The parabolas in the same diagram represent Gerber's parabolic relation. In the region lying between the straight lines *OA*, *OB* the stress

changes sign during the cycle. On the right of this region the stress is always tensile; on the left of OB the stress is always compressive.

Suppose that in the case of a steel having an ultimate tensile strength of 39·4 tons per in.², the safe range of stress for reversed stresses is 14·8 tons per in.². To find the safe range

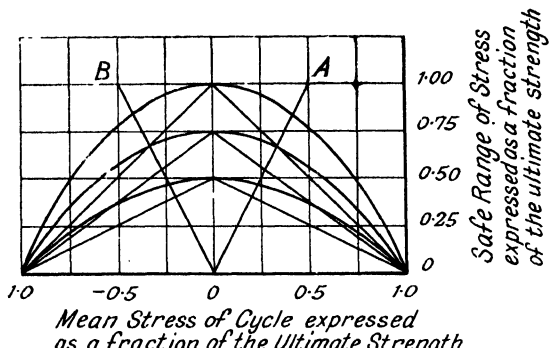


FIG. 168. DIAGRAM ILLUSTRATING MODIFIED GOODMAN LAW

of stress by the Goodman law when the mean stress is 18 tons per in.² we have

$$R' = 2 \times 14{\cdot}8 = 29{\cdot}6 = \frac{29{\cdot}6}{39{\cdot}4} f = 0{\cdot}752 f$$

$$S = 18 = \frac{18}{39{\cdot}4} f = 0{\cdot}457 f$$

The safe range may be found from the graph, or more simply by the formula

$$\begin{aligned} \mathrm{R} &= \mathrm{R}'(1 - \mathrm{S}/f) \\ &= 0{\cdot}752 f(1 - 0{\cdot}457) = 0{\cdot}341 f \\ &= 13{\cdot}45 \text{ tons per in.}^2 \end{aligned}$$

Relative Endurance of Mild and H.T. Steel. An instructive way of considering the effects of cycles of different ranges of stress is due to Haigh.

Suppose a stress cycle has a mean value S = 5 and an alternating value A = 7. If these values be plotted on a diagram having values of S for abscissae and values of A as ordinates we obtain a single point P on the diagram. For the

relation S = A we obtain a straight line inclined at 45° to the axes. This line is shown dotted in Fig. 169. When the plotted point *P* lies above this line the stress cycle representing it is necessarily reversing in character since A is greater than S. When, however, the point lies below the line the corresponding stress cycle does not reverse.

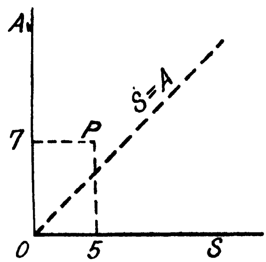


FIG. 169. HAIGH'S DIAGRAM

Referring to Fig. 170, a graph can be drawn through all points representing combinations of stress that will just cause fatigue after very long endurance. Another graph can be drawn to represent more severe cycles that will cause fatigue after one million reversals. A further graph may be plotted to represent combinations of stress that are severe enough to cause immediate plastic strain in the material.

To construct such a diagram, the plastic line *yy* is drawn at an angle of 45° through the points *yy* which represent the yield point as determined by a simple tensile test of the material. The equation of the line is S + A = *y*.

To draw the fatigue curve it is necessary to conduct several series of tests with different values of A to ascertain the fatigue

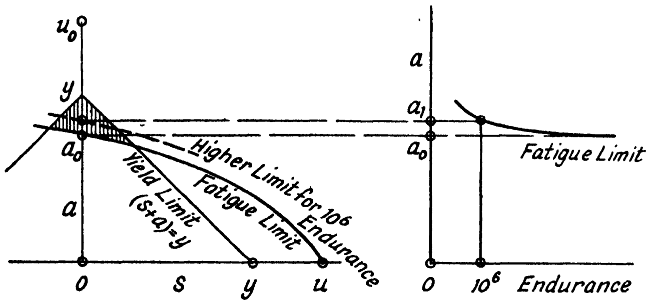


FIG. 170. SHOWING COMBINATIONS OF STRESS THAT WILL PRODUCE FAILURE BY FATIGUE

limit for one selected value of S. The endurance curve for each series is plotted as in the figure on the right and the results combined as in the left-hand figure.

It will be seen the fatigue line cuts off a part of the

right-angled triangle that would enclose all safe combinations of stress if plastic yield alone were in question, and the safe zone in the diagram is seen to be bounded partly by the fatigue line and partly by the yield line.

The serious risk of fatigue arises when a working stress cycle is represented by a point that lies within the shaded zone shown in the figure.

A serviceable formula for the fatigue graph is given as

$$a_s = a_0[1 - k_1(S/u) - k_2(S/u)^2]$$

where a_s = the fatigue strength when the stress s is acting,

a_0 = the ordinary fatigue limit when $s = 0$,

u = the ultimate tensile strength.

k_1 and k_2 are constants for the particular metal in question.

Figs. 171 and 172 illustrate particular cases.

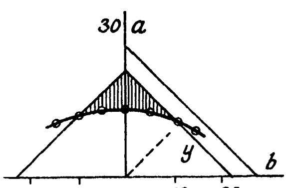


FIG. 171. RESULTS OF TEST ON MILD STEEL

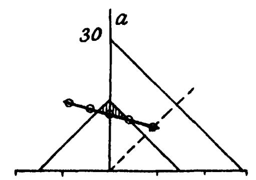


FIG. 172. RESULTS OF TESTS ON NAVAL BRASS

The first is an example of Gerber's parabolic relation. The material is mild steel for which $k_1 = 0$ and $k_2 = 1$.

The second example refers to a naval brass, tested in the "as rolled" condition. Professor Haigh remarks that this series of tests was probably the first that ever revealed any difference between the action of push and pull in fatigue. In this metal, as in many others, pull tends to reduce the fatigue limit while push increases the resistance to fatigue. Here $k_1 = 1$ and $k_2 = 0$.

The k_2 coefficient is of little importance in practice as its effect is not marked until the steady stress is a fairly high proportion of the ultimate strength. On the other hand the k_1 coefficient makes its influence felt with small values of the steady stress S, and hence is of some practical importance. Steady stresses tend to reduce the fatigue limit when they act in

tension and to raise the fatigue limit when they act in compression.

The diagram can be used to illustrate the contrast between steels of different types as regards safety under repeated stresses. Fig. 173 (*a*) is representative of mild steel. Suppose the ultimate strength to be 30 tons per in.^{2}, the yield point 21 tons per in.^{2} and the fatigue limit 18 tons per in.^{2} The fatigue line is drawn as a parabola, $k_1 = 0$, $k_2 = 1$.

The conditions here are favourable to immunity from fatigue which can only occur when the steady stress is so small that the

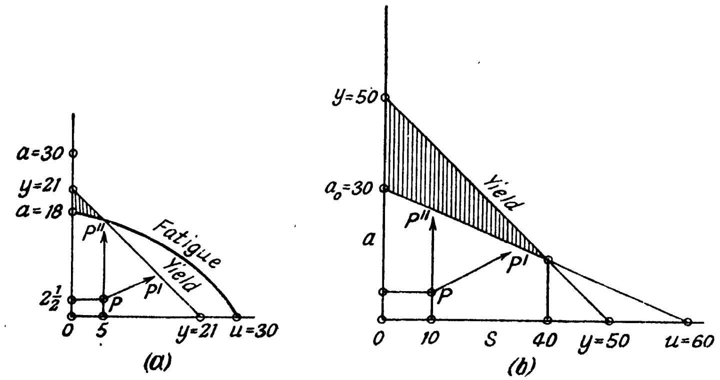


FIG. 173. DIAGRAM ILLUSTRATING THE RELATIVE SAFETY OF MILD AND ALLOY STEELS

point representing the combination falls within the small shaded area above the fatigue line, and persists for a sufficiently long period to produce a crack. In any other circumstances the metal fails by yielding.

The other diagram, Fig. 173 (*b*), represents the very different relations that commonly prevail with high tensile steel. The ultimate strength is assumed to be 60 tons per in.^{2} with a fatigue limit of 30 tons per in.^{2} The fatigue line is straight, with $k_1 = 1$, and $k_2 = 0$.

The dangerous zone, represented by the shaded area, is seen to be relatively much larger than in the previous example. With steel of this kind fatigue can occur under any value of steady stress up to 40 tons per in.^{2}, and further, a large part of the fatigue zone is so wide that fatigue can be produced rapidly by excessive ranges of alternating stress well above the fatigue limit but below the limit for plastic yield.

TABLE XV

VALUES OF THE COEFFICIENTS k_1 AND k_2 FOR VARIOUS STEELS

Material	Tensile Strength Tons per in.^{2}	Yield Point Tons per in.^{2}	Ordinary Fatigue Limit Tons per in.^{2}	Coefficients		Ratio Fatigue Limit / Tensile Strength
				k_1	k_2	
Mild steel	25·2	21·0	13·0	0	1	0·516
Naval brass	28·7	14·5	12·0	1	0	0·418
3½ per cent nickel steel	55·0	48·5	26·8	0	1·47	0·487
3½ per cent nickel steel	49·8	40·6	26·8	0	2·26	0·537
3½ per cent nickel steel	45·3	28·9	22·0	1·37	0	0·485
3·6 per cent nickel steel	51·3	34·7	23·0	1	0	0·448
0·29 per cent nickel steel ann.	32·3	17·4	13·0	0·37	0	0·403
Cast iron	14·1	—	4·5	1·6	0·6	0·320
Wrought iron	22·8	14·5	7·1	0	1	0·311

(*Reproduced by permission of the Institution of Automobile Engineers.*)

The Schenk Pendulum Fatigue Testing Machine. In the Schenk Pendulum Fatigue Testing Machine, Fig 174, the specimen is tested under four-point loading, but the load is applied by means of a pendulum or beam *A* provided with a sliding weight *B*. The pendulum is pivoted at C and is linked to the free ball bearings through the bridle *D*. Oil dashpots are fitted in order to damp out vibrations. Deflection indicators are also fitted and are so proportioned that with a standard specimen a movement of the indicator needle of 1 cm. corresponds to a stress of 2 kg. per mm.^{2} (1·27 ton per in.^{2}) at the periphery of the specimen. As the specimen tends to heat up when under test it is enclosed in a box through which oil can be forced for cooling. A split thermocouple may be clamped to the middle of the test piece, the connecting wires being brought to insulated rings mounted one on each spindle. Small brushes rubbing on the rings enable connection to be made to the millivoltmeter or other temperature indicator for measuring the temperature rise.

To ascertain the energy absorbed through hysteresis effects, precision instruments are provided for measuring the energy input to the driving motor. The energy absorbed by the test bar is found by subtracting the no-load energy of the machine from the total energy.

The method of testing is based on the principle that in the region of the fatigue limit a significant enlargement of the hysteresis loop is apparent. If the energy absorbed be plotted against the maximum stress in the specimen the graph bends

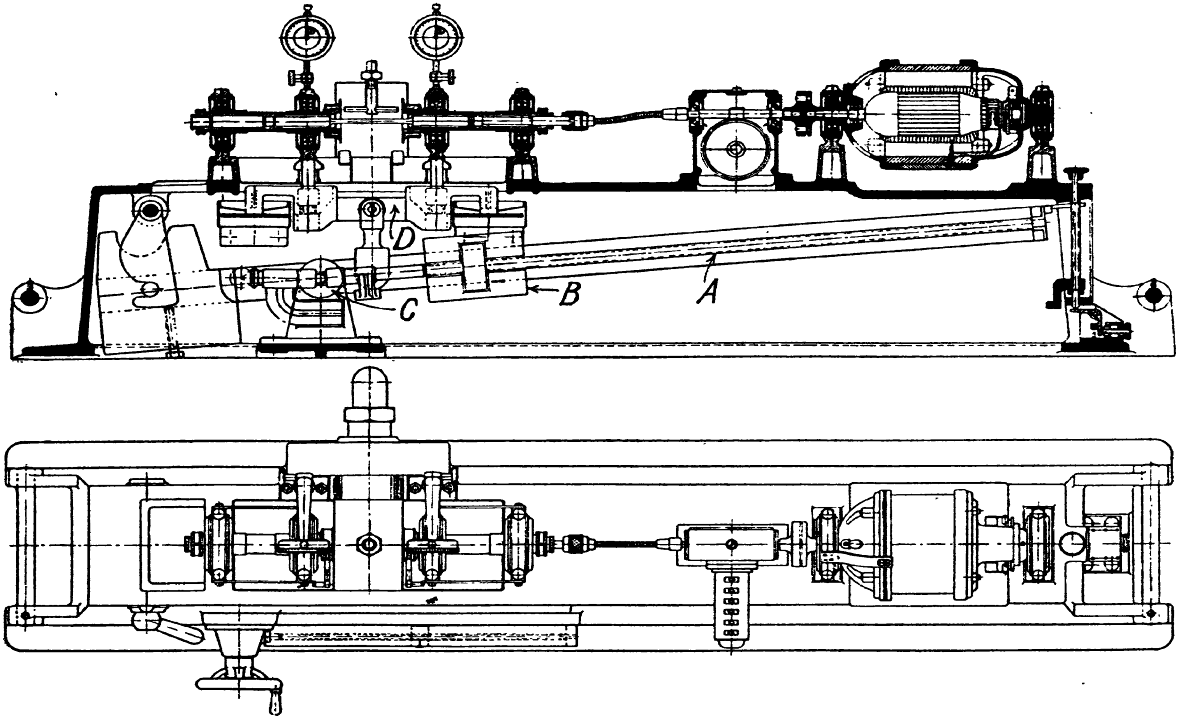


FIG. 174. SCHENK PENDULUM FATIGUE TESTING MACHINE
(Carl Schenk. Darmstadt)

very suddenly and turns through an angle approaching 90°. In Fig. 175 curve 1 shows the total energy absorbed per unit volume per cycle as the stress is increased, while curve 2 represents the energy absorbed by the test piece.

The point A at which the curve begins to bend is referred to as the *first characteristic point.* The energy curve rises steeply after this point is passed and a tangent drawn to the curve to intersect the axis of abscissae yields the point B, the *second*

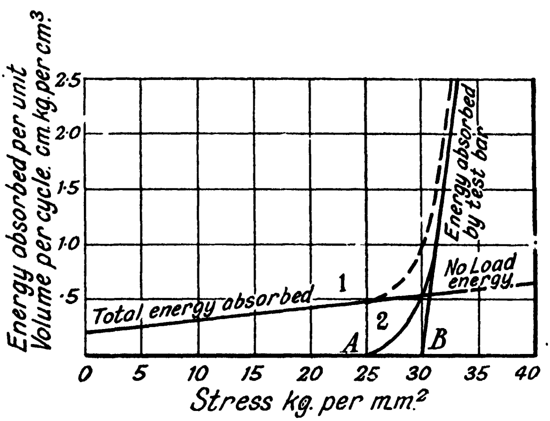


FIG. 175. ILLUSTRATING HYSTERESIS-LOSS METHOD OF TESTING

characteristic point, which for ferrous materials is assumed to indicate the fatigue limit.

The curve of temperature rise is similar to the energy absorption curve.

With non-ferrous materials the fatigue limit falls near to the point A.

Characteristic curves for steel and brass are given in Fig. 176 (a) and (b).

To use the method for short-time fatigue tests comparison should first be made with results obtained by the usual duration tests on material similar to that which it is proposed to test.

Fatigue Tests by Torsional Oscillations. Another machine by Schenk, of Darmstadt, for making fatigue tests employs the method of torsional oscillations. It is shown diagrammatically in Fig. 177. The relation between the applied torque and the angle of twist is indicated on a ground glass screen by an optical method. The test bar forms part of an oscillating system

consisting of a flywheel counterweight at one end and a smaller mass at the other.

The torsional oscillations are effected through an electro-

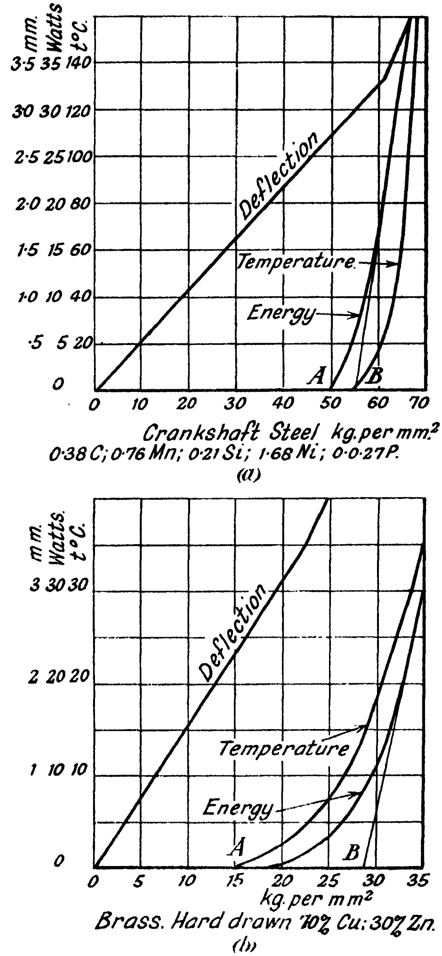


FIG. 176. CURVES OF ENERGY LOSS AND TEMPERATURE RISE

magnetic exciting system. A beam of light from the lamp is projected successively on to two groups of mirrors and thence to the glass screen, where it appears as a point of light. The

position of the spot of light relative to the axes indicates the values of both torque and twist. During a cycle the point of light traverses a closed curve which, owing to the high frequency of the oscillations, appears as a stationary hysteresis loop. The loop which reduces to a straight line when little or no power is absorbed, becomes increasingly inflated as more and more power is absorbed by the test piece. The area of the loop is a measure of the energy absorbed per cycle.

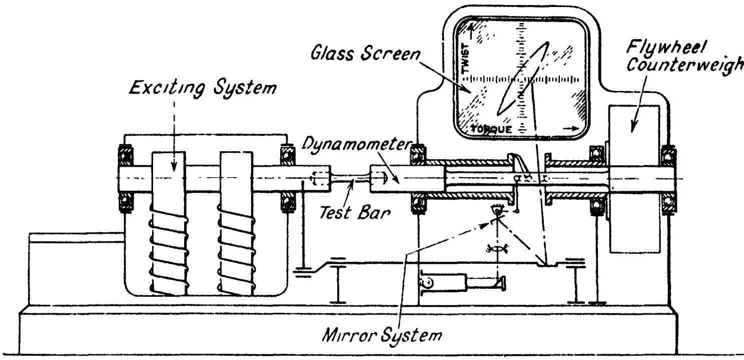


FIG. 177. SCHENK FATIGUE TESTING MACHINE EMPLOYING THE METHOD OF TORSIONAL OSCILLATIONS

Short-time Fatigue Tests. The chief objection to the adoption of the fatigue test in commercial testing is the length of time required to obtain an accurate determination of the fatigue limit. In view of this the search for a short time fatigue test has been prosecuted vigorously. Among the various tests that have been proposed are—

(*a*) Tests based on inelastic action or on mechanical hysteresis.
(*b*) Tests based on rise of temperature.
(*c*) Tests based on deflection of the specimen.
(*d*) Tests based on the energy absorbed per cycle.
(*e*) Tests based on change of electrical resistance as fatigue cracks develop.

The test described on p. 221 is intended to provide a short time method.

Recently, a test based on repetition of stress combined with tensile tests on the same specimens has been put forward by Moore and Wishart and termed the *overnight* test. Five or

six test pieces are taken and the Rockwell hardness of each determined.

The specimens are then subjected to about 1 400 000 cycles of stress in a rotating bar machine so as to cover a range of values on both sides of the estimated fatigue limit. After testing in this way the specimens are removed from the machine and pulled in ordinary tension. If a specimen breaks before completing the standard number of cycles its tensile strength for the purpose of plotting is regarded as zero.

The results of the tension tests are corrected to Rockwell hardness and the results plotted against the magnitude of the reversed stress. The resulting curve will, in general, show a maximum or greatest value of the reversed stress which is taken as giving the fatigue limit.

The following results are taken from a test by Moore and Wishart—

Magnitude of Reversed Stress Applied for 1 400 000 cycles lb. per in.^{2}	Rockwell Hardness Number, B Scale	As Tested lb. per in.^{2}	Tensile Strength after 1 400 000 cycles reduced to Rockwell Hardness Scale of 60 lb. per in.^{2}
26 000	70·0	61 500	61 500
28 000	61·2	62 400	61 200
30 000*	60·0	63 200	63 200*
31 000	60·0	62 000	62 000
32 000	60·2	62 000	62 300
33 000	61·2	0†	

* The fatigue limit is thus 30 000 lb. per in.^{2}

† Specimen broke before completing 1 400 000 cycles.

The results of other tests are shown in the graphs in Fig. 178.

The theory on which the test is based is that under repeated stressing materials tend to increase in strength, due presumably to cold work, while on the other hand they suffer loss of strength through fatigue cracks or otherwise. The beneficial influence predominates below the fatigue limit and also to some extent above that limit, but here fatigue cracking leads to ultimate failure.

Gough's view is that there is no reason why any short-time test should be expected to prove reliable, and that a dividing line at the fatigue limit above and below which material behaves differently does not exist, and further, the variability in commercial material may be sufficient to mask effects brought out by such a test as that proposed above.

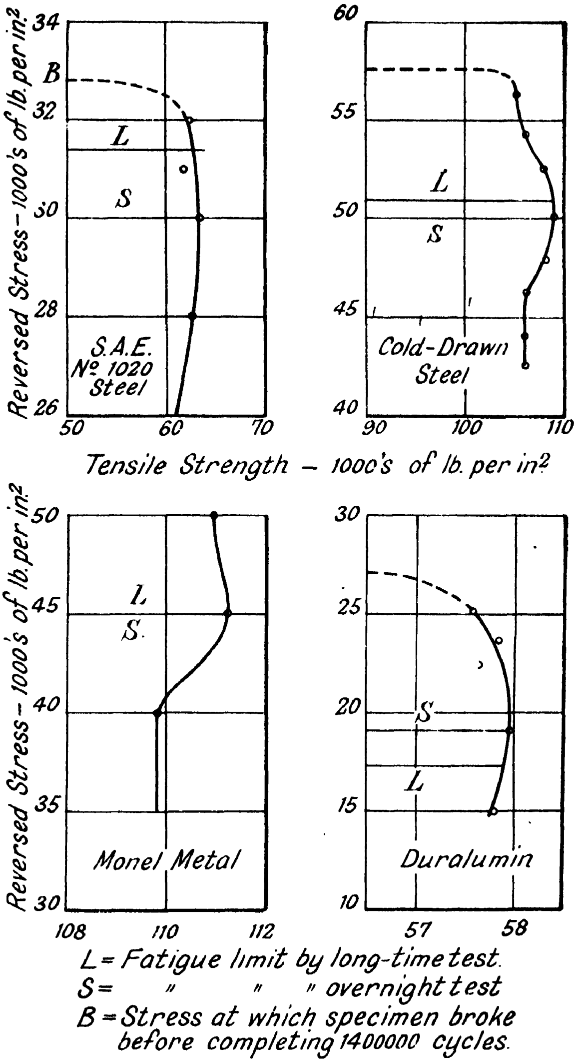


FIG. 178. RESULTS OF FATIGUE TESTS BY THE "OVERNIGHT" METHOD

(*American Society for Testing Materials*)

High Frequency Fatigue Tests. G. N. Krause, using an air turbine as a driving motor, has constructed a rotating-bar machine to run at 30 000 r.p.m. Comparison of tests on various steels, cast iron, brass and duralumin, with tests of the same materials in a machine of the ordinary type running at 1 500 r.p.m., shows that there is no appreciable speed effect up to 10 000 r.p.m. At 30 000 r.p.m., the fatigue limit is raised, in some instances substantially.

Fatigue tests on wires at speeds up to 2 000 vibrations per sec. were carried out by C. F. Jenkin prior to 1925. The method consisted in passing an alternating current through a length of wire supported at its nodes in a magnetic field. If the length of wire and the frequency of the current are suitably adjusted, the wire will vibrate at its natural frequency in its first mode; or, as it is termed, as a "free-free" bar.

Although in Jenkin's experiments comparatively thin wires were employed, no such limitation is imposed, and the method has, moreover, the very real advantage that specimens may be used with the minimum of preparation.

A commercial instrument embodying a development of the principle is the G.E.C. electro-magnetic fatigue tester made by Salford Electrical Instruments, Ltd.

The apparatus, which is designed to accommodate specimens 18 in. long and $\frac{1}{2}$ in. diameter, consists of two main units; the *vibrator unit* and the *power unit* together with a synchronous clock or a logarithmic recorder. The operation may be followed by reference to the schematic diagram, Fig. 179.

Under each end of the test bar is a magnet system carrying d.c polarizing coils and a.c exciting coils. Under the centre of the bar is an electro-magnetic pick-up. The tuned relay and the auxiliary relay are mounted on the vibrator unit and the remainder of the apparatus indicated in the diagram is contained in the power unit.

The alternating current from the mains passes via the switch to the rectifier. This is of the grid-controlled type, enabling the d.c. output and so, ultimately, the amplitude of vibration of the bar, to be adjusted to the required value. The d.c. output is fed to the d.c. polarizing coils and to the d.c.–a.c. inverter. The a.c. output of the inverter is fed into the exciting coils and a condenser is provided to neutralize the inductance.

The effect of the magnets on the test bar thus takes the

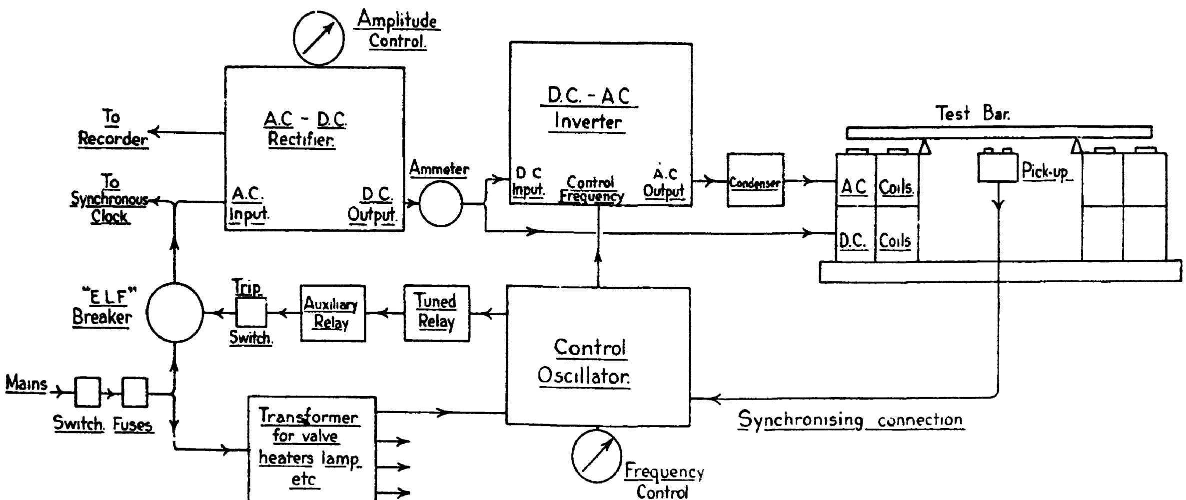


FIG. 179. SCHEMATIC DIAGRAM OF G.E.C. ELECTR -MAGNETIC FATIGUE TESTER

(G.E.C.)

form of a steady force due to the d.c. polarizing current plus a superimposed alternating force due to the alternating current. If the frequency of the applied a.c. is sufficiently near to the natural frequency of the bar the latter will vibrate. The frequency of the inverter output is determined by the control oscillator. This is provided with a frequency control by means of which the frequency can be adjusted so that the bar vibrates. When this happens, the vibration induces a small voltage in the pick-up, and this voltage, being fed back to the control oscillator, causes the oscillator to lock into synchronism with the vibration of the bar.

In the tuned relay, a flat steel spring loaded in the centre is stretched over a small magnet supplied from the control oscillator. The tension in the spring is adjustable so that the natural frequency of the spring can be adjusted to a slightly lower value than the natural frequency of the test bar. When the test piece begins to crack, its frequency falls; the control oscillator being synchronized, its frequency also falls and the tuned relay comes into operation. The spring vibrates and touches a contact which operates the auxiliary relay. This closes a contact which operates the trip coil of the circuit-breaker, thus shutting down the apparatus. The supplies to the synchronous clock and recorder are interrupted at the same time, thus providing a record of the time of stopping. The trip mechanism can be disconnected by means of the trip switch when initial adjustments are being made.

The tuned relay is provided with a dial calibrated in cycles per second, indicating the frequency to which it is tuned. It can therefore be used for measuring the natural frequency of the bar for the determination of E in addition to its normal purpose of tripping the apparatus.

The amplitude of vibration is measured optically. A small gold bead is mounted on a clip which is attached to the centre of the test piece. The bead is illuminated by a 6-volt lamp which provides a point of light. The point is drawn out into a line when the bar vibrates. The length of the line is a measure of the amplitude and is observed in a low-powered microscope. The amplitude can be measured to about 0·0002 in.

Specimens other than of standard form may be tested by suitable adaptation. Non-magnetic specimens require a steel armature at each end and a steel clip opposite the pick-up.

The maximum displacement and bending moment curves

are shown in Fig. 180. The distance between the nodes is $0{\cdot}552 \times$ length of bar.

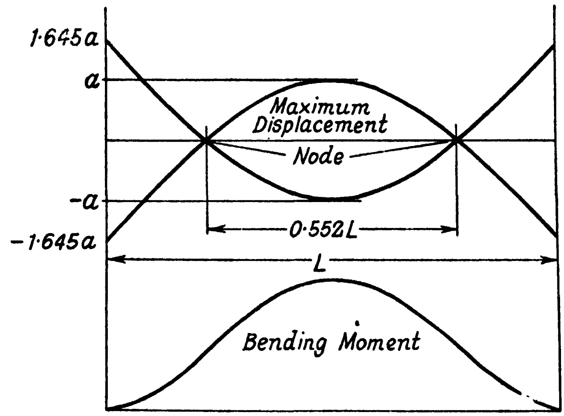


FIG. 180. HIGH-FREQUENCY FATIGUE TESTS: MAXIMUM DISPLACEMENT AND BENDING MOMENT CURVES

With the notation—

y = deflection from mean position;
x = distance from mid-point of bar;
L = length of bar;
d = thickness or diameter of bar;
a = amplitude at mid-point of bar;
E = Young's Modulus;
ρ = density of material;
f = frequency (cycles per sec.);
I = moment of inertia of cross section of bar;
k = radius of gyration of cross section of bar;

the equation of motion for a uniform "free-free" bar, vibrating in its first mode is

$$y = a[\mathrm{A} \cos m(x/\mathrm{L}) - \mathrm{B} \cosh m(x/\mathrm{L})] \cos \omega t,$$

where

$$\omega = 2\pi f = (m^2 k/\mathrm{L}^2)\sqrt{(\mathrm{E}g/\rho)},$$

and

A, B and m are constants having the respective values $\mathrm{A} = 1{\cdot}1532$, $\mathrm{B} = 0{\cdot}1532$, $m = 4{\cdot}7300$.

The bending moment is given by

$$EIa(m/L^2)\,[A\cos m(x/L) + B\cosh m(x/L)]\cos\omega t.$$

For standard bars—

Maximum bending moment $= 29{\cdot}39\ EIa/L^2$.
Maximum stress $= 14{\cdot}61\ Eda/L^2$.

The value of Young's modulus may be determined by measuring the frequency of the vibrating bar. The amplitude should be kept small while making this test.

For standard bars Young's modulus is given by the formula—

$$E\ (\text{lb./in.}^2) = 388 \times \text{weight of bar (lb.)} \times \text{frequency}^2.$$

When testing specimens of non-magnetic material or of non-uniform section, suitable modifications must be made to the above formulae.

Strength under Combined Alternating Stress. Since materials in practice, are subjected more commonly than otherwise to the effects of a complex system of stresses—particularly to combinations of flexural and torsional stresses—it is obviously desirable to determine their fatigue strength under combined alternating stress rather than to rely on tests made under bending or torsion alone.

A machine for this purpose has been developed at the National Physical Laboratory. The test piece is loaded by the inertia forces of revolving out-of-balance weights, the inertia of the remaining vibrating and rotating parts being eliminated from the specimen by a suspensory spring system whose natural frequency is tuned to that of the working speed of the machine.

Fig. 181 shows the general arrangement. One end of the test piece *S* is rigidly clamped in a movable bracket *K* bolted to a casting *G* which is attached to the baseplate *B*. The other end of the specimen is held in a collar *C* to which the arm *A* is pivoted about a vertical axis passing through the centre of the specimen. The disc *D* carrying the out-of-balance weights *W* is supported on an axle *F* suitably clamped to the ends of the springs *E* which are rigidly held in a bracket carried by the baseplate.

The disc is driven through a belt drive by a synchronous motor *M* running at 1 500 r.p.m. The bracket *F* is connected by links *L* and the arm *A* so that the deformation of the test piece shall not be seriously constrained. The out-of-balance forces developed at the axle are transmitted as an alternating

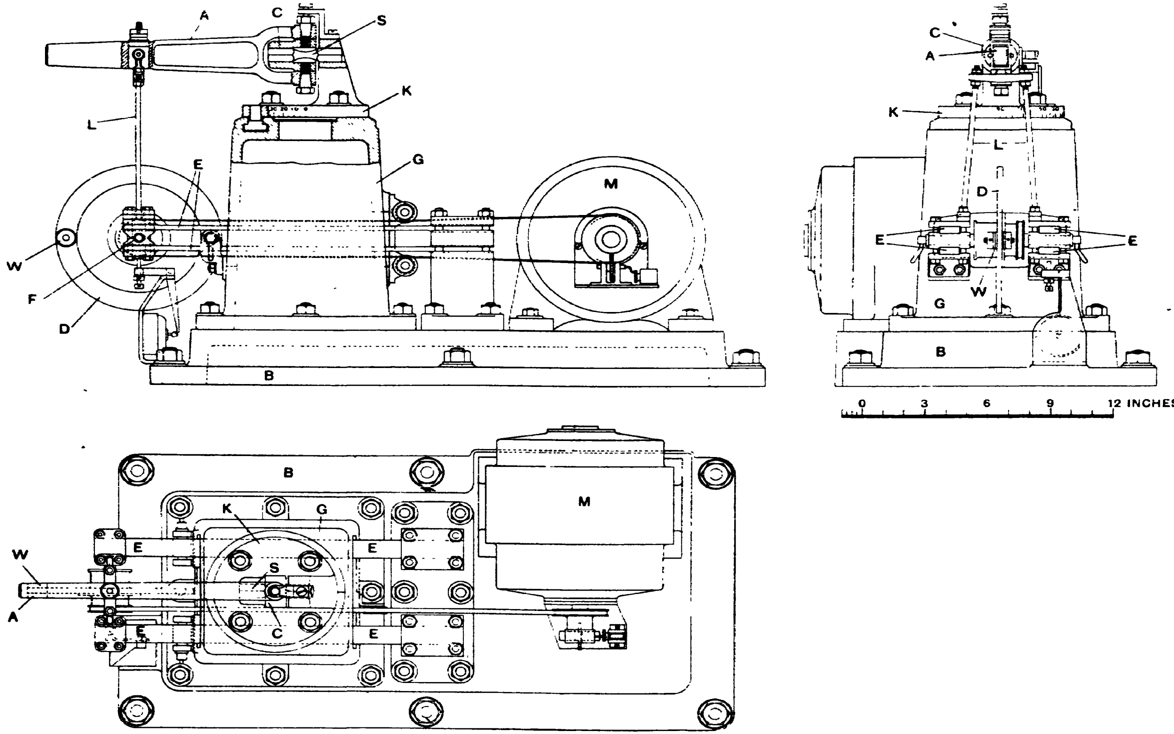


FIG. 181. COMBINED STRESS (BENDING AND TORSION) FATIGUE TESTING MACHINE
(Institution of Mechanical Engineers)

moment through the arm A. By varying the ratio of the pulley diameters the speed of the disc may be adjusted to the resonant frequency of all the moving parts vibrating on the outer ends of the springs. This adjustment is effected by the use of a very flexible specimen in the form of a thin strip.

In the position shown in Fig. 181, cycles of reversed plane bending stress are imposed on the specimen. The bracket K and the collar C can be rotated and clamped in any other position so that the arm makes an angle θ with the longitudinal axis of the test piece. The bending moment applied is then proportional to $\cos \theta$ and the torque to $\sin \theta$. The bending moment and torque are in phase so that both reach their maximum and minimum values simultaneously.

When $\theta = 0°$, cycles of reversed plane bending are applied and when $\theta = 90°$, cycles of reversed torsion are applied, while any desired combination is obtained by the appropriate choice of θ.

The out-of-balance weights which generate the alternating stresses are attached at two diametrically opposite positions on the disc; any weights not in use are housed on the arm A over the centre line of the links L, the mean weight distribution of the moving parts being thereby maintained constant in all tests. By suitable choice of out-of-balance weights, ranges of bending stress on a standard specimen (0·3 in. diameter) varying from $\pm$ 5 to $\pm$ 50 tons per in.^{2} may be obtained.

When it is desired to make an independent static calibration of the machine, the links L are removed, thus disconnecting the loading arm A from the springs, etc. A specimen, reserved for calibration purposes, is then placed in the machine and loaded by applying a series of weights to a scale pan suspended from the loading arm A at the point at which the links L are normally connected. The deflection of the specimen is then determined by measuring, on the screen of a specially constructed camera box, the distance traversed by a spot of light reflected from the mirrors N and O. In this way a static load-deflection curve is obtained which can be compared with the curve obtained under running conditions, using appropriate balance weights.

CHAPTER XI

ELASTIC CONSTANTS: TESTING OF WIRE AND SHEET METAL

Modulus of Elasticity of Wires. The elastic constants of material in the form of wire and sheet metal are often conveniently determined by indirect methods.

For the direct determination of Young's modulus for a wire, Searle's apparatus is available. (Fig. 182.) Two wires A, B, eight to ten feet long, are suspended vertically from a beam and at their lower ends carry brass stirrups which support the ends of a sensitive spirit-level L. The level is supported by a pivot in one stirrup and by the point of a micrometer screw in the other. The wire carrying the pivoted end of the level is simply a suspension wire and is kept taut by a weight W attached to the stirrup. The other wire is the test wire. In commencing a test the level is first adjusted to the horizontal so that the bubble is at the centre of its run. A 1 lb. weight is placed on the hanger carried by the test wire and by means of the micrometer screws the level is again brought to the horizontal position. The difference between the micrometer readings gives the extension of the test wire. Readings are taken for 8 or 9 loads and a load-extension curve plotted. The procedure for determining Young's modulus is similar to that described on page 87. The extension of the wire can be measured to an accuracy of 0·01 mm., or approximately to 0·002 mm. by estimation.

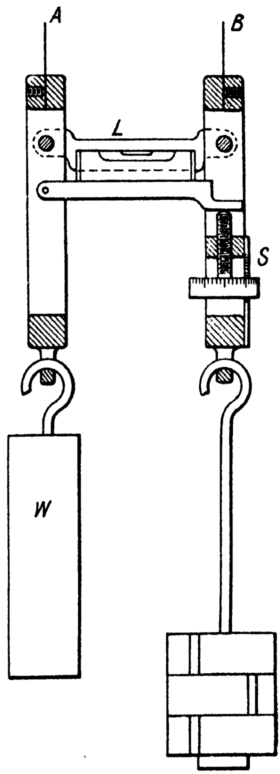


FIG. 182
SEARLE'S APPARATUS FOR DETERMINING YOUNG'S MODULUS OF WIRES

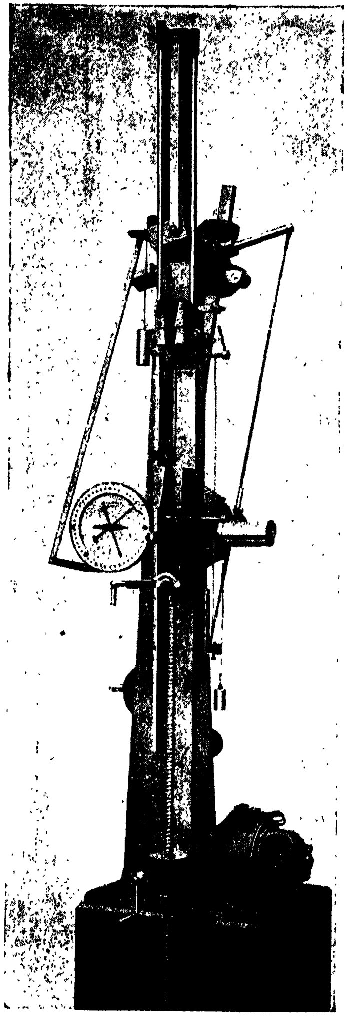


FIG. 183. AMSLER WIRE TESTING MACHINE
(*Alfred J. Amsler & Co.*)

Connecting links prevent one stirrup from twisting relatively to the other about a vertical axis but permit relative motion of the stirrups in a vertical direction. The apparatus is made by Messrs. W. G. Pye & Co., Cambridge.

The Amsler Wire Testing Machine. For thick wires some form of extensometer such as the Westinghouse described on page 132 may be used, the load being applied by a wire tester. The Amsler machine is shown in Fig. 183. The upper gripping head is suspended from a lever attached to the inner end of the pendulum spindle. The inclination of the pendulum, which is a measure of the load, is transmitted by a set of parallelogrammatic rods to a pointer moving over a graduated dial. One arm of this parallelogram moves a horizontal rod which engages a small pinion on the axle of the pointer and moves the pointer proportionally to the load.

A 1 000 lb. machine has ranges of 1 000, 500, 200, and 100 lb. The various ranges are obtained by removing or adding weights to the pendulum or by altering the length of the pendulum rod.

To measure the elongation of the specimen use is made of a vertical rod having a $\frac{1}{20}$ in. graduation. If the variation in the distance between the two gripping heads is to be regarded as giving the elongation of the test wire, the rod is fastened to the lower gripping head by a clamp, so that during the test it

moves downwards with the head. Parallel to this rod is a small wire fixed to the frame of the machine. A tube provided with an index slides along this wire. During the test the index is pushed downwards by the upper gripping head and remains stationary when the specimen fractures. The index thus indicates on the graduation of the rod the variation in the distance between the gripping heads, and this corresponds closely to the elongation of the specimen.

If the actual elongation of the specimen is required over a fixed gauge length an extensometer must be used. The machine is provided with an autographic recorder. For a load of 1 000 lb. the accuracy of measurement is ½ per cent of the maximum of the respective range from half the range upwards. This is absolute, that is, on the 1 000 lb. range the accuracy is 5 lb.; on the 500 lb. range, 2½ lb., and on the 200 lb. range, 1 lb.

Modulus of Rigidity by Torsional Oscillations. The value of the shear modulus N of a wire, may be found by means of a torsion tester of the type described on page 144, or by the method of oscillations.

A metal bar, termed an *inertia bar*, is suspended by a vertical wire several feet long (Fig. 184). If a slight angular displacement in the horizontal plane be given to the bar the wire acts as a spring, and when the bar is released it will oscillate to and fro for some time.

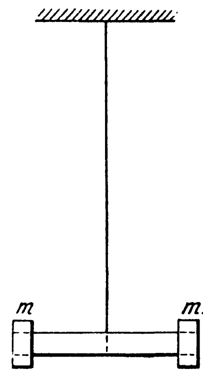


FIG 184
ILLUSTRATING THE METHOD OF TORSIONAL OSCILLATIONS

If l be the length of the wire, d its diameter, and N the modulus of rigidity of the material, the couple which the wire exerts on the bar for a small angular displacement θ, from the position of equilibrium is

$$T = \pi N\theta d^4/32l$$

The periodic time of an oscillation is given by

$$\tau = 2\pi\sqrt{\frac{\text{angular displacement}}{\text{angular acceleration}}}$$

$$= 2\pi\sqrt{\frac{\text{angular displacement}}{\dfrac{\text{torque}}{\text{moment of inertia}}}}$$

If I represents the moment of inertia we have then,

$$\tau = 2\pi\sqrt{\frac{\theta}{T/I}}$$

and substituting for T,

$$\tau = 2\pi\sqrt{(32lI\theta/\pi N\theta d^4)}$$

whence it is seen that θ cancels and the periodic time is independent of the angular displacement. For this to hold, however, the angular displacement and the restoring force must be proportional throughout the range.

If the moment of inertia of the bar is not known it may be calculated or it may be determined as follows.

Two small masses m, m are attached to the ends of the bar equidistant from the axis of the wire. If the masses are of small dimensions their radius of gyration may be taken to be the same as the distance r of the centre of each mass from the axis of the wire. Then, approximately, their combined moment of inertia about this axis is $I_1 = 2\,mr^2$.

The bar is given a slight displacement and the time of an oscillation is found by noting the time for 30 complete oscillations. The time of oscillation is determined similarly when the bar carries the masses m, m.

Since $T_1^2 \propto I$ in the first case, and in the second $T_2^2 \propto I_1 + I$, the ratio of the squares of the periodic times is thus

$$\tau_1^2/\tau_2^2 = I/(I + I_1)$$

Hence $$I = \tau I_1/(\tau_2^2 - \tau_1^2)$$

Having found the moment of inertia of the bar and the periodic time, the modulus of rigidity can be calculated from the formula

$$N = 120lI\pi/d^4\tau^2$$

Instead of a bar a circular disc may be used, with a thin annular ring for the added mass.

The moment of inertia of a rectangular bar of length l, breadth b, and depth d about a vertical axis through its c.g. parallel with the depth is

$$\text{mass} \times [(b^2 + l^2)/12]$$

For a round bar of length l and diameter d,

$$I = \text{mass}(d^2/16 + l^2/12)$$

For a circular disc of diameter d about an axis perpendicular to its plane $I = \text{mass}\ (d^2/8)$.

Searle's Dynamical Method of Determining Young's Modulus. A dynamical method, due to Searle, may be employed to determine Young's Modulus.

The ends of the wire are soldered into two clamping screws which are secured to two equal bars AB and CD, Fig. 185. Two light hooks are screwed into the bars at G and G' so that the hooks are perpendicular to the wire. The system is suspended by two parallel threads at least 20 in. long attached to the hooks. If the bars are turned through equal angles ϕ in opposite directions and then set free the system will execute harmonic vibrations in the horizontal plane. If the vibrations are small the effects of the horizontal and vertical displacements of the centres of the bars may be neglected.

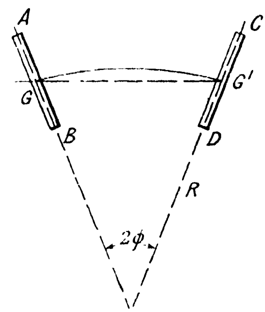


Fig. 185. Illustrating Method of Determining Young's Modulus by Bending Oscillations of a Wire

For all practical purposes the forces on the system reduce to equal couples exerted by the bars on the wire which is bent to the arc of a circle.

The bending moment is EI/R where E is Young's modulus, I the moment of inertia of the section of the wire about an axis through its centroid and R the radius of curvature to which the wire of length l is bent. $I = \pi d^4/64$ in.^{4} and from the figure $R = l/2\phi$.

If I_D is the moment of inertia (dynamical) of either bar about a vertical axis the angular acceleration of the bar is

$$\alpha = \frac{\text{couple}}{\text{moment of inertia}} = \frac{EI}{I_D R} = \frac{2EI\phi}{I_D l}$$

and the period of oscillation

$$\tau_E = 2\pi\sqrt{(I_D l/2EI)}$$

Example: Consider a steel wire 0·047 in. diameter and 15 in. long. Moment of inertia of the bar $I_D = 13$ lb.-in.^{2} units. Average time for one complete oscillation 1·188 sec. The value of E is required in lb. per in.^{2} so $g = 32{\cdot}2 \times 12$ in. per sec. per sec.

Hence $$I_D = \frac{13}{32 \cdot 2 \times 12}$$

and $$E = \frac{128 \times 13 \times 15 \times \pi}{(1 \cdot 188)^2 \times 32 \cdot 2 \times 12 \times (0 \cdot 047)^4}$$

$$= 29 \cdot 65 \times 10^6 \text{ lb. per in.}^2$$

In carrying out this experiment the amplitude of vibration must be small; not much greater than 3°. The ends of the two bars should be tied together in the constrained position by a piece of thread and vibration started by burning the thread. A pointer should be set close to the end of one of the bars to assist in finding the time of vibration. Several measurements of the diameter should be made at various points along the wire.

The modulus of rigidity may be found as already described, page 237, by clamping one bar to a support and noting the time of oscillation of the suspended bar. Then

$$N = 128\pi I_D L / \tau_N^2 d^4$$

The ratio of E to N is simply

$$E/N = \tau_N^2 / \tau_E^2$$

which allows Poisson's Ratio σ to be calculated (page 14.)

$$\sigma = E/2N - 1$$

Values of the elastic constants of several materials in the form of wire are given in the following table—

TABLE XVI

ELASTIC CONSTANTS OF WIRES

(G. F. C. Searle)

Material	E lb. per in.²	N lb. per in.²	σ
Carbon steel . . .	$28 \cdot 72 \times 10^6$	$11 \cdot 41 \times 10^6$	0·258
Brass (hard drawn) . .	14·81	5·392	0·376
Phosphor bronze . .	17·40	6·320	0·378
Copper (annealed) . .	18·73	4·083	0·608
Nickel (hardened) . .	34·78	10·76	0·614
*Platinoid . . .	19·70	5·217	0·887
†German silver . .	16·72	3·795	1·207

* 54 Cu. 24 Ni. † Cu. Ni. Zn. Alloy.

From the relation $3K(1 - 2\sigma) = 2N(1 + \sigma)$ we see that if $\sigma > \frac{1}{2}$ either K or N would be negative. The explanation is that the material of the last four wires is not isotropic.

Searle's Method of Determining the Elastic Constants of Strip Metal. Metal in the form of strip can be tested in tension and the value of Young's modulus found in the usual way. The modulus may also be found by loading a thin strip of metal as a beam. The simple theory of bending then fails and Poisson's ratio enters into the relation between deflection and bending

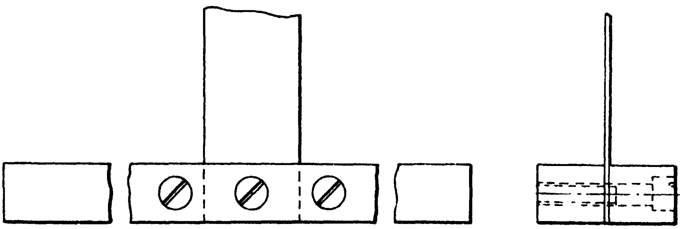


Fig. 186. Inertia Bars Clamped to Strip

moment. Because of this the formula connecting Young's modulus and the bending moment is not, in itself, sufficient for the determination of E from a series of observations of load and deflection, and it is necessary, in addition, to find the modulus of rigidity by an oscillation method.

A convenient size of specimen for the purpose is a strip about 24 in. long $\times$ 2 in. wide. A hole is drilled at each end and the strip firmly clamped between bars as shown in Fig. 186. One pair of bars is fixed firmly to a suitable support so that the strip hangs vertically and the lower bars are allowed to oscillate in a vertical plane. The procedure is similar to that described on page 237.

If l be the length of the strip,
$2a$ the width of the strip,
$2t$ the thickness of the strip,

the relation between the angle of twist and the twisting couple is

$$T = 16Nat^3\phi/3l$$

If L is the length of each bar, A its width and M its mass the moment of inertia of the two bars about the axis of the strip is, by the theorem of parallel axes,

$$I_D = \tfrac{2}{3}M(L^2/4 + A^2) + 2M(A/2 + t)^2$$

Neglecting the second term we have approximately the value

$$I_D = \tfrac{2}{3}M(L^2/4 + A^2)$$

The angular acceleration is T/I_D, that is $16Nat^3\phi/3lI_D$.

Hence if τ be the periodic time

$$N = 3lI_D\pi^2/4at^3\tau^2 \text{ lb. per in.}^2$$

The relation between E and the bending moment M required

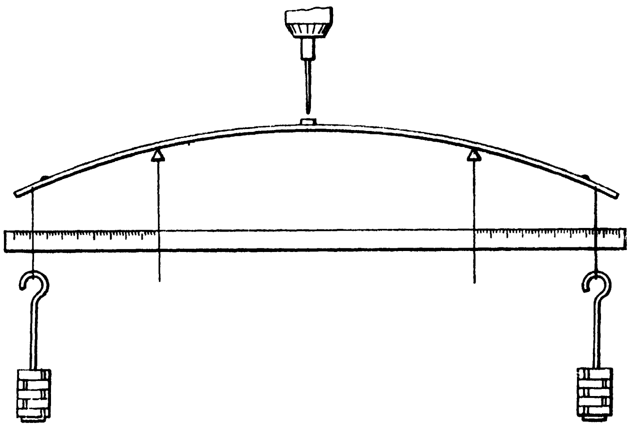


FIG. 187. METHOD OF SUPPORTING AND LOADING STRIP DURING DEFLECTION TEST

to bend the strip so that the longitudinal filaments have a radius of curvature R is

$$M = 4at^3E/3(1 - \sigma^2)R$$

In performing the bending test the strip is supported on knife-edges, Fig. 187, which should be spaced so that the part of the blade between them is as nearly as possible straight when unloaded. This distance is approximately half the length of the blade. The deflections may be measured by means of a pointed micrometer forming part of an electric circuit arranged to light a small bulb when contact occurs. Or the curvature may be determined by the mirror method to be described later.

Small hangers are suspended by threads attached to the ends of the beam with seccotine. Readings are taken with several

loads on the hangers. The radius of curvature is calculated from the formula

$$R = l^2/2h + h/2$$

where l is the distance between the supports and h the deflection of the central point.

If W is the load on each scale pan and p the average distance of the hangers from the supports

$$E/(1 - \sigma^2) = 3WRp/4at^3$$

The mean value of $E/(1 - \sigma^2)$ is obtained by plotting a graph.

Substituting for E from $E = 2N(1 + \sigma)$

we have

$$2N(1 + \sigma)/(1 - \sigma^2) = 3WRp/4at^3$$

therefore

$$\sigma = 1 - 8Nat^3/3WRp$$

Substituting this value of σ in

$$E/2N = (1 + \sigma)$$

we have

$$E = 4N(1 - 4Nat^3/3WRp)$$

The theory of the foregoing method is given fully in Searle's *Experimental Elasticity*.

Determination of Poisson's Ratio by the Method of Flexures. The change of slope of a bent beam is best observed by means

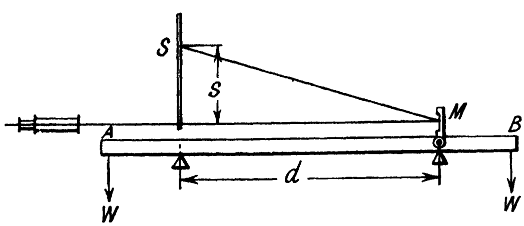


Fig. 188. Method of Measuring the Slope of a Beam

of a telescope and scale. A plane mirror M is attached to the beam immediately over a knife-edge, and a scale S and a telescope are arranged as in Fig. 188, the ray of light from the mirror to the telescope being horizontal and the scale vertically over the knife-edge support.

If d is the distance between the knife-edge supports and if

the tangent to the beam at the knife-edge turn through a small angle and give a displacement on the scale of amount s for a load W applied at A and B then the radius of the arc to which the beam is bent is

$$R = d^2/s$$

If h is the deflection of the mid point of the beam we have from the geometry of a circle $R = d^2/8h$. Comparison with the previous result shows that the movement over the scale is eight times the deflection of the beam.

Greater accuracy is obtained if the distance between the scale and the mirror be increased and the curvature calculated from the change of slope.

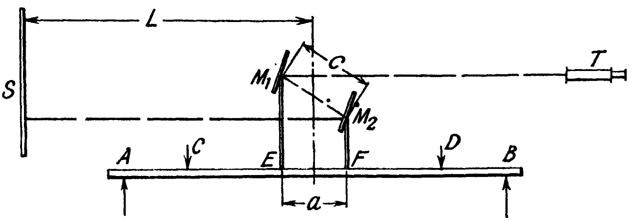


FIG. 189. CARRINGTON'S METHOD OF MEASURING FLEXURE

Carrington's method of determining E and σ by the method of flexures is as follows—

The beam is supported at A and B as in Fig. 189 and loaded at C and D. Two mirrors M_1 and M_2 are attached to small columns fixed to the beam at E and F. The distance EF in Carrington's experiments was 1 in. Observations being taken by means of a telescope and scale, the modulus of elasticity may be calculated from the formula

$$E = \frac{24a(L + c/2 + a/2)}{bt^3} \cdot \frac{M}{s}$$

where

b is the breadth of the beam,
t the thickness,
a the distance between the mirror columns,
c the distance between the mirrors, and
L the scale distance.

To determine Poisson's Ratio the lateral curvature must be

measured. The mirrors are arranged as in Fig. 190. If the dimensions L, a and c are the same as in the previous experiment it is only necessary to plot curves connecting bending moment and scale reading for both cases, when the ratio of the slopes of the resulting straight-line graphs will be the value of Poisson's Ratio.

In any case Poisson's Ratio is given by

$$\frac{\text{Longitudinal curvature}}{\text{Lateral curvature}}$$

The thickness of the beam must not be too great to prevent a measurable degree of lateral curvature under moderate loads.

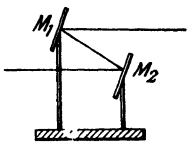


FIG. 190. CARRINGTON'S METHOD OF MEASURING LATERAL FLEXURE

Poisson's Ratio may be found by measurement of the longitudinal and lateral strains in a test bar under tension, the lateral strains being measured by means of a delicate extensometer, as for instance Coker's extensometer described on page 122.

In a recent investigation, Lyse and Godfrey employed a split collar of spring steel clamped on to a tension specimen by set screws and giving a magnification of approximately two at the gap. The amount of gap opening was measured by means of a Huggenberger extensometer placed across the gap. Lateral strains were measurable to 0·000005 in. for a 1 in. diameter specimen.

These investigators determined Poisson's Ratio by several methods and found that for structural steel σ varied from 0·271 to 0·302, while for alloy steels the ratio varied between 0·272 and 0·320.

Commercial Tests on Wire. Commercial tests on wire comprise tensile, bending, twisting and wrapping tests. Tensile tests are conveniently made in a machine of the type described on page 236. Some British specifications require that a lever type machine be used. The sample of wire must not be straightened or in any way prepared before testing.

The specified rate of application of the load varies with the material being tested. Nine-tenths of the minimum breaking load must be applied quickly and the remainder at a steady rate until fracture of the wire occurs. With galvanized stay wire, for instance, the time to be occupied in applying the final

tenth of the load is given as 20 sec., the total time from the application of the load to the break being approximately 30 sec.

Twisting tests are made by clamping the ends of the test wire in suitable grips, one of which can be rotated, while the other, although prevented from rotating, is free to slide longitudinally. (See Fig. 108.) Small hand machines are usually employed for this test. The twist is applied through a geared handwheel and a counter registers the total number of turns up to fracture.

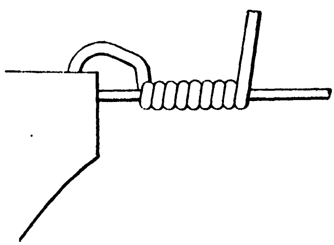


Fig. 191. Wrapping Test

Specifications usually call for a minimum number of twists in a given length without fracture, the number of twists to be determined by first making an ink mark on the untwisted wire and then counting the number of spirals in the gauge length. The full number of twists must be visible between the grips. The speed of testing should not exceed one revolution per second.

For hard drawn copper wire the specified number of twists ranges from 15 to 30 on a length of 6 in., the number depending on the diameter of the wire.

The wrapping test is made by gripping the wire in a vice and wrapping one portion of the wire around the other as shown in Fig. 191.

The wire is wrapped six times around its own diameter in the same direction, unwrapped, and again wrapped in six turns in the same direction as the first wrapping. In some cases only one wrapping is specified.

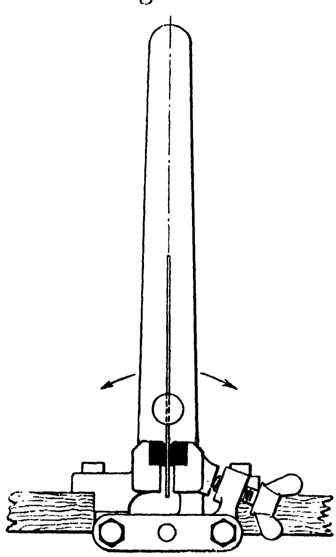


Fig. 192. Hand Bending Machine for Wires

Bending tests are made by means of an apparatus shown in Fig. 192, the wire being bent to and fro through 90° or 180° over a specified radius. For phosphor bronze wire the angle of

bend specified is 90°, and for wires of 0·032 in. to 0·16 in. diameter the radius of bend varies from $\frac{1}{16}$ in. to $\frac{3}{16}$ in. The minimum number of bends varies from eleven to six, the larger number corresponding with the thinner wires.

In continental practice the wires are put in tension by means of a spring to aid in forming the wire to the required radius. The method of gripping the wire between flat jaws is open to objection.

Fatigue Tests on Wire.* Wire in service is often subjected to repeated stressing as in winding and haulage ropes, and failure through fatigue is not infrequent. Ordinarily, the testing

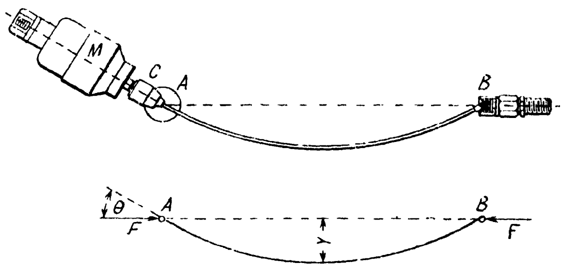


FIG. 193. ILLUSTRATING PRINCIPLE OF MACHINE FOR MAKING FATIGUE TESTS ON WIRES

of wire in fatigue is attended with somewhat formidable difficulties, but a machine recently invented by Haigh and Robertson, and made by Messrs. Bruntons (Musselburgh) Ltd., overcomes most, if not all, of the difficulties encountered by an investigator when attempting such tests.

The principle of the machine is illustrated in Fig. 193. One end of the test wire A is gripped in a chuck C as in the headstock of a lathe and the other end B is arranged to run on a simple form of ball thrust bearing that serves as a tailstock. The tailstock bearing is screwed forward to flex the wire and the chuck C, together with its driving motor, swivels about a vertical axis on bearings A. The use of the ball-bearing B and the vertical axis of swivelling ensures that the wire bends in the horizontal plane under the action of equal and opposite forces F acting along the line AB.

The bending moment and curvature of the wire vary along its length and are greatest at mid-span. The sample of wire is

* See page 228.

kept rotating until it eventually breaks by fatigue. Fracture occurs at or near mid-span and not near the chuck.

The wire does not "whirl" about the straight line AB but rotates about its own curved axis of flexure. The speed of operation is about one million cycles per hour.

When the angle of flexure is small the theory of the test corresponds to that of the Euler strut (page 21). The curve assumed by the centre line of the wire is approximately

$$y = Y \sin(\pi x/\mathrm{L})$$

where

Y = deflection at the mid-point,

L = length of sample,

y = deflection at a distance x from one end.

The inclination and curvature are given by

$$i = dy/dx = (\pi Y/\mathrm{L})\cos(\pi x/\mathrm{L}) = \theta \cos(\pi x/\mathrm{L})$$

$$1/r = d^2y/dx^2 = -(\pi^2 Y/\mathrm{L}^2)\sin(\pi x/\mathrm{L})$$

$$= -(\pi\theta/\mathrm{L})\sin(\pi x/\mathrm{L})$$

θ is the inclination at the end, as indicated by a vernier on the swinging headstock of the machine.

The greatest curvature is at the mid-point and is given by

$$1/\mathrm{R} = \pi\theta\mathrm{L}$$

where R denotes the corresponding least radius of curvature.

The bending strain at the mid-point of a wire of diameter d is

$$e = d/2\mathrm{R} = (\pi/2)\theta(d/\mathrm{L})$$

and the bending stress

$$f = \mathrm{E}e = (\pi/2)\theta(d/\mathrm{L})\mathrm{E}$$

where E is Young's modulus.

When the angle θ is greater than about 20° the curve of flexure becomes the "Elastica" and the foregoing results need slight correction.

In addition to the bending stress a small compressive stress acts in the wire. This may be calculated by using Euler's formula, but generally the compressive stress may be ignored.

The end thrust applied to the test piece and the grip of the

chuck affect the whirling critical speeds. These several speeds —which are to be avoided—are proportional to

$$\frac{\sqrt{(E/\rho)}}{d(L/d)^2}$$

where ρ is the density of the metal. In general it is desirable to run between the second and third criticals.

Test pieces are usually 150 diameters long and lengths of from $3\frac{1}{2}$ to 30 in. may be tested.

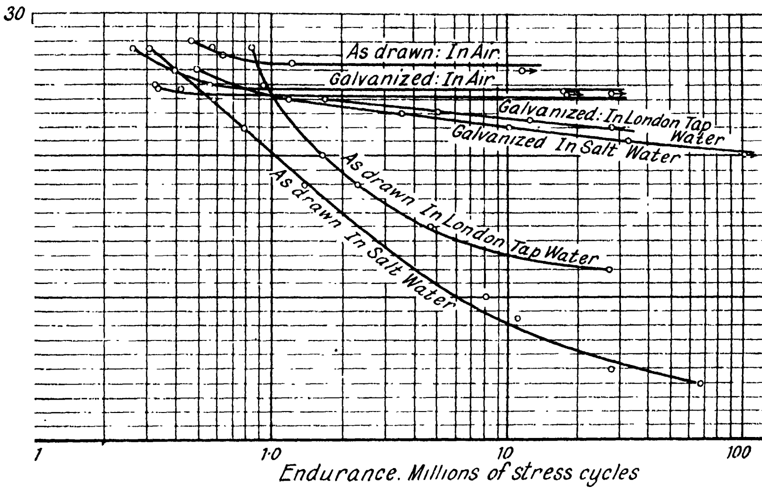


FIG. 194 ENDURANCE CURVES FOR STEEL WIRES UNDER CONDITIONS OF CORROSION FATIGUE
(*Bruntons (Musselburgh) Ltd.*)

Some endurance curves for steel wire under various conditions of corrosion fatigue are given in Fig. 194.

Tests of Sheet Metal. The mechanical tests made on sheet metal comprise tensile tests, bend tests, reverse bend tests, hardness tests, and a cupping test.

TENSILE TESTS. Various forms of test piece are called for in specifications. Two forms only have been standardized by the British Standards Institution for material not exceeding 0·128 in. (10 S.W.G.) in thickness.

The "short" form has a gauge length of 2 in., is $\frac{1}{2}$ in. wide, and has a parallel length of $2\frac{1}{2}$ in. The "long" form, intended to be used in cases where the general elongation of the test piece

is likely to be more informative than the local elongation, has a gauge length of 8 in., is $\frac{3}{4}$ in. wide, and has a parallel length of 9 in.

The radius of the transition curve at the ends of the straight portion is 1 in. The preparation of the test piece is somewhat difficult especially from very thin sheets. A combined filing and drilling jig in which the specimen can be dressed to shape is of great assistance, as axial loading of the test piece is essential.

For measuring extensions the extensometer employed should possess a sensitivity of 1/20 000 in. For elastic measurements a

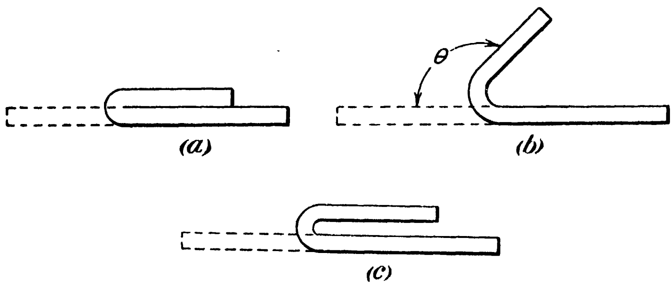


FIG. 195.* BEND TESTS FOR SHEET METAL

form of mirror extensometer may be used, or a pair of Huggenberger extensometers, clamped one on each side of the specimen. If measurements are to be carried into the plastic region a coarser form of instrument must be employed. A form of direct-reading instrument has been designed to meet the necessary requirements, and a diagram of this is given in the B.S.S. No. 485.

With some materials the speed of testing is not without influence on the result, but so far it has been found impossible to specify generally rates of testing.

BEND TESTS. In bend tests the bending is caused by pressure or by a succession of blows. The metal is bent as in Fig. 195, (*a*), (*b*), and (*c*), being closed on itself or bent through a specified angle. The test as in (*a*) is known as the *hammer* or *seaming* test. Tinplates usually stand this test when the line of bend is "across the grain," i.e. at right angles to the direction of rolling,

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but not infrequently fail when bent "with the grain," i.e. parallel with the direction of rolling.

Considerable differences of opinion exist as to the value and fairness of this test.

The test is sometimes made less severe by bending the sheet through an angle round a specified radius as in (*b*). The angle may reach 180° as in (*c*).

The test piece must remain in contact with the former during the bending operation and if, after removing the constraint, the test piece assumes a slightly different shape this is to be ignored. Specimens are deemed to be satisfactory if no cracks show on the convex side.

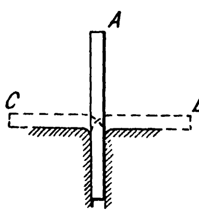

Fig. 196.* Reverse Bend Test

Reverse Bend Tests. A strip of the material *A*, Fig. 196, is clamped in a vice provided with rounded jaws and is then bent

(*a*) through 90° into position *B*, then back to *A* and then to and fro between these positions; or

(*b*) is first bent into position *B*, then through 180° into position *C* and then alternately into positions *B* and *C* until a crack appears.

The number of bends to fracture is counted, the first bend through 90° being ignored, although in the tinplate trade it is counted as half a bend.

Some constraint is needed to ensure contact between the test piece and the jaws of the vice. If this condition be not fulfilled the radius to which the strip bends will generally differ appreciably from the radius of the jaw. In the Jenkins Bend Tester, an improved form of the machine shown in Fig. 192, a strip about 2 in. long and ½ in. wide is gripped between jaws which are rounded to a radius of 0·04 in. This radius is used for all sheets up to 0·4 mm. For sheets up to 2 mm. a larger machine is used with jaws of 0·08 in., 0·125 in. and 0·25 in. radius. A hand-lever working on a floating fulcrum is provided with a hardened steel roller which catches the projecting strip and permits it to be bent to and fro. Pressure is applied to the roller

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by means of a spring so that in the course of making a bend the test piece is forced to conform with the curvature of the jaw.

The length and width of the strip appear to be without influence on the results provided the strips are cut from the sheet with a close-set sharp pair of shears; otherwise the burred edges tend to crack prematurely.

With a given material the number of bends which a test

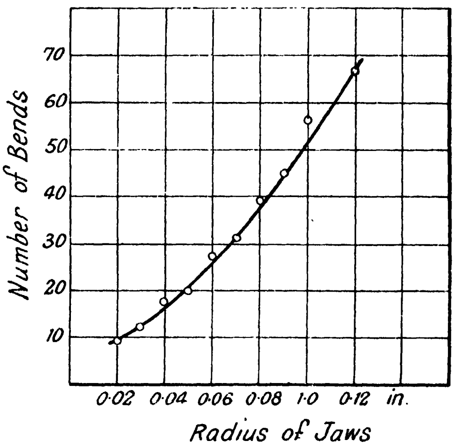


FIG. 197. INFLUENCE OF RADIUS OF BEND ON NUMBER OF BENDS TO CAUSE FRACTURE
(*J. G. Godsell*)

piece will withstand will depend on the direction in which it has been cut from the sheet.

(See Bibliography p. 309.)

It has been established that if

n_0 = the number of bends sustained by a strip taken from from the direction of rolling.

n_{90} = the number of bends sustained by a strip perpendicular to the direction of rolling,

the number of bends n_θ that a strip cut at an angle θ to the direction of rolling will withstand is given by

$$n_\theta = \sqrt{(n_0^2 \cos^2 \theta + n_{90}^2 \sin^2 \theta)}$$

In the application of this formula the initial bend through 90° is counted as half a bend.

With mild steel sheets the rate of bending appears not to affect the result, but whether this holds for all alloys has not been definitely established.

The influence of the radius of the bend is shown in Fig. 197 which is plotted from some tests of annealed mild steel sheets

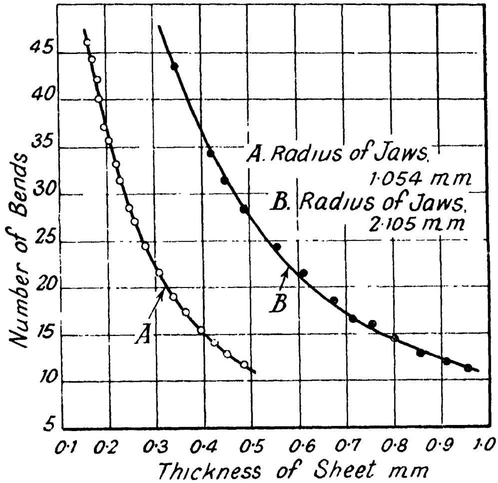


FIG. 198. INFLUENCE OF THICKNESS OF SHEET ON NUMBER OF BENDS TO CAUSE FRACTURE

(*J. G. Godsell*)

0·34 mm. thick. Each plotted point represents the mean of 10 tests, the strips being bent through 180° and the first 90° bend counted as half a bend. The effect of the thickness of the sheet with a given radius of jaw is shown in Fig. 198.

If the number of bends be plotted against the ratio

$$\frac{\text{Radius of bend}}{\text{Thickness of strip}} = \frac{R}{T}$$

a straight line is obtained having the equation

$$N = K(R/T - 0{\cdot}8)$$

in which K depends on the material.

Logarithmic plotting leads to the result $N = C(R/T)^{\frac{2}{3}}$ for

mild steel, where C depends only on the quality of the material. Fig. 199.

TABLE XVII

STANDARD FIGURES FOR TINPLATE AND BLACKPLATE
(For use with the Jenkins Alternating Bend Tester (0·040 Jaws)

NUMBER OF BENDS REQUIRED TO FRACTURE

Thickness		*Substance	Ordinary Coke		Deep Stampers	
			Weak Way	Strong Way	Weak Way	Strong Way
in.	mm.					
0·0155	0·395	135	4½	7½	5	8½
0·0146	0·371	127	5½	9½	6	10½
0·0124	0·315	108	7	12	8	13½
0·0116	0·295	100	7½	13	8½	15
0·0109	0·277	95	7½	13	8½	15½
0·0104	0·264	90	8½	15	9½	16½
0·0098	0·248	85	9½	16½	11	19
0·0092	0·234	80	10½	18	—	—
0·0086	0·218	75	11½	20	—	—
0·0081	0·206	70	12½	22	—	—
0·0074	0·188	65	13½	24	—	—

CUPPING TEST. The object of the cupping test is to ascertain the ductility of the material. The test piece, which is a disc about 3 in. diameter cut from the sheet, is placed between two steel rings of rectangular section and enclosed in a box or frame. A punch having a rounded nose is placed in contact with the disc and is forced down by means of a screw and nut or by compression in a small testing machine. The load, and the depth of the cup when fracture is observed, provide a measure of the ductility of the metal.

In Continental practice the disc is gripped firmly between the rings, but in this country and America clearance is allowed so that the test piece can draw down with but little restriction. In the form of test known as the *Erichsen Test* the internal diameter of the ring is 27 mm. and the radius of the nose of the punch 10 mm. Practice differs slightly in different countries.

The standard Erichsen machine is shown in Fig. 200. In

* Weight in lb. of a standard box of tinplates, i.e. 112 sheets 14 in. × 20 in. = 31 360 in.^{2}

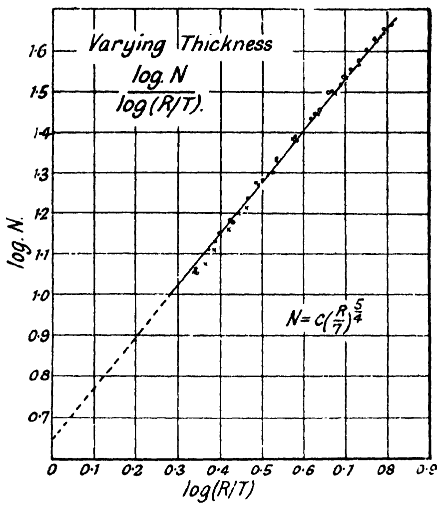


FIG. 199. RELATION BETWEEN NUMBER OF BENDS TO FRACTURE AND THE RATIO RADIUS OF BEND-THICKNESS OF SHEET
(*J. G. Godsell*)

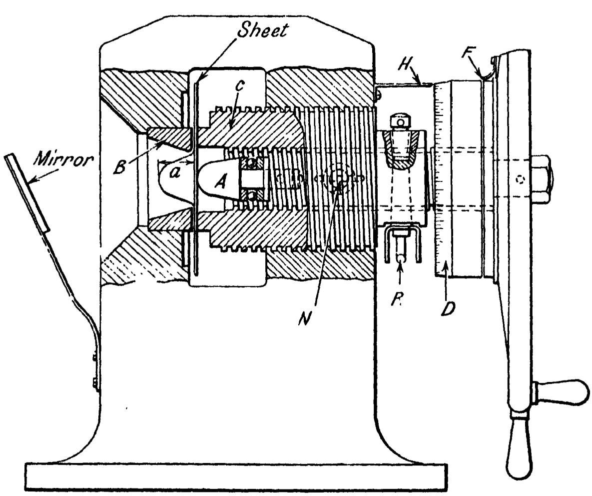


FIG. 200. STANDARD ERICHSEN MACHINE FOR CUPPING TEST

making the test the scale *D* is set to zero by shifting the movable collar on which it is engraved until the spring *F* snaps into a small hole in the collar. The specimen is inserted between the die *B* and the holder *c* and the handwheel turned until the specimen is firmly clamped. The thickness can then be read off from the scale *D*. The total range of this scale is 5 mm. and readings can be continued on scale *H*.

After the thickness of the sheet has been noted the handwheel is turned back five divisions on scale *D* ($\frac{5}{100}$ mm.) in order to give the test piece a small amount of play. The holder *c* is secured in this position by means of the wing nut *N*. The scale on *D* is now moved until its zero coincides with that of *H*. The gear is changed by pressing against the milled ring *R* and the handwheel turned clockwise. The tool *A* now moves forwards and bulging of the sheet is noted in the mirror. The image in the mirror is carefully watched until fracture occurs, when the depth *a* of the impression is read off on the scale. The rate of testing should be reduced as the point of fracture is approached, in order to obtain an exact reading.

This type of test seems to be regarded as a method of detecting unsuitable material, but one that is less discriminating than the tensile or alternating bend tests where more satisfactory material is being dealt with. The test certainly provides valuable information regarding the probable surface appearance of a finished pressing, this being related to the grain size of the material.

In one respect the test appears to be superior to the tensile test, in that it tests the ductility of the sheet in all directions. Factors disadvantageous to the test from a quantitative aspect are the uncertainty of the frictional effects at the surface in contact with the punch, the amount of drawing at the grips, and the determination of the exact point at which fracture commences.

However, the cupping test is largely used in the routine testing of metals, and some curves of Erichsen values are given in Figs. 201A, 201B, 201C, and 201D. The curves, with one or two exceptions, are not the Erichsen Standard Trade Quality Curves.

The depth of cup at fracture will not itself provide a true index of the deep drawing qualities of a material, but consideration must be given to the type of fracture and the appearance of the dome. The fracture should be circumferential in metals which are required for drawing operations. Metal which

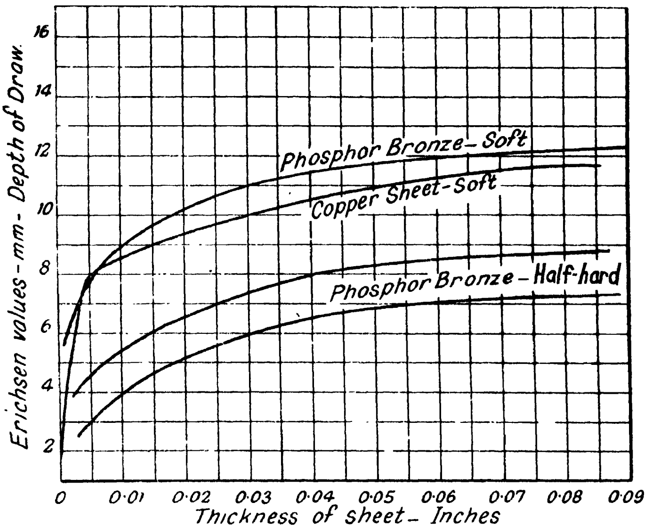


FIG. 201 A. CURVES OF ERICHSEN VALUES

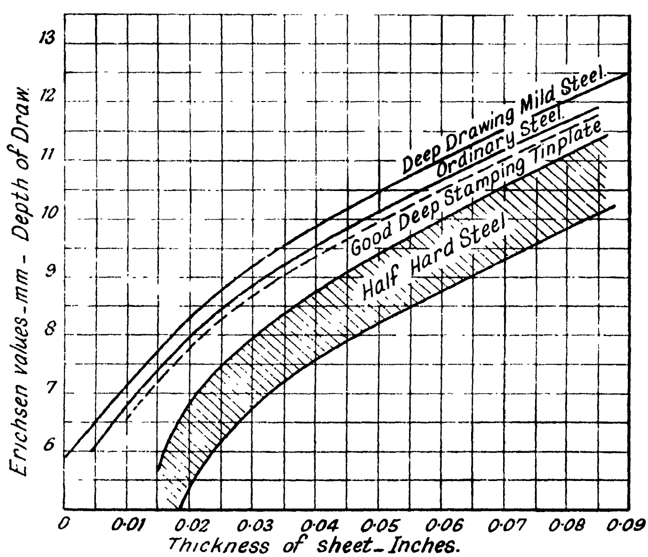


FIG. 201 B. CURVES OF ERICHSEN VALUES

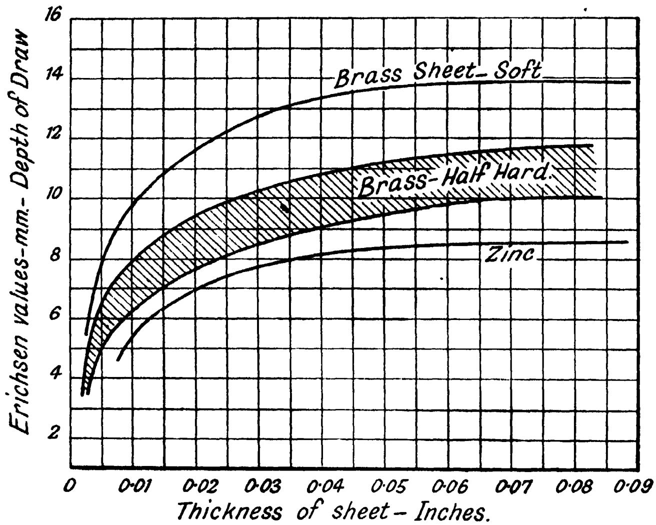


FIG. 201C. CURVES OF ERICHSEN VALUES

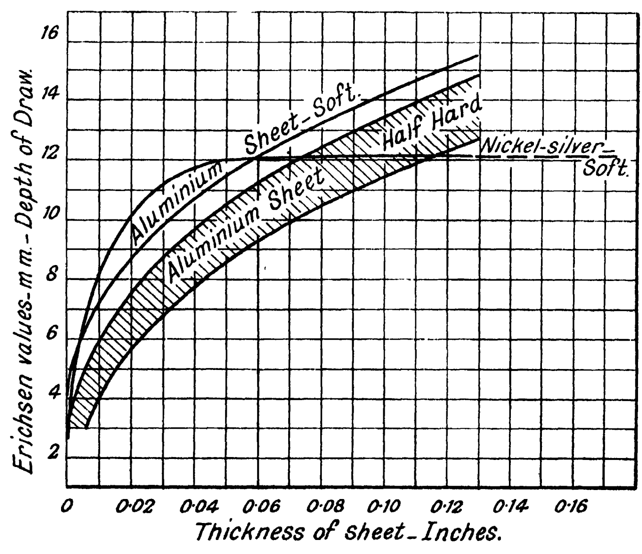
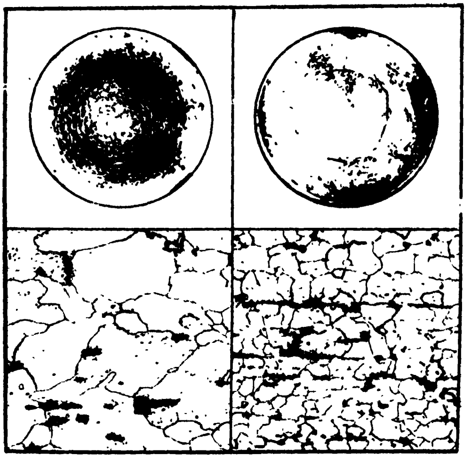


FIG. 201D. CURVES OF ERICHSEN VALUES

has been reduced considerably by cold rolling, such as hard nickel-silver, will give a fracture in one direction only and will not be suitable for folding and drawing. A rough or crinkled dome indicates a loose or coarse structure that will offer increased resistance to the drawing tools, and may result in premature breakage even though the metal is soft.

With non-ferrous metals and alloys, Armco iron, and electro-deposited iron, the roughness is invariably due to over-annealing, but this is not true in the case of dead mild steel sheet.



(a) (b)

FIG. 202. APPEARANCE OF DOME IN MILD STEEL

(a) Rough dome—coarse grained structure.
(b) Smooth dome—same material commercially re-annealed.
(*J. G. Godsell*)

Abnormal grain growth, and therefore a rough Erichsen dome, in the case of dead mild steel sheet, is due to under-annealing after cold work. On annealing above Ac_3, i.e. 900° C., the structure, no matter how coarse, is refined and is not appreciably coarsened at any temperature below 1 200° C. Good normalized mild steel exhibits a smooth dome (Fig. 202). A dome close grained in appearance is usually only encountered in non-ferrous materials and is generally caused by oxidation of the metal during annealing or by excessive pickling. In all cases the domes should be smooth, and deep-drawing qualities should not possess a value falling below the respective curves shown. Half-hard materials should fall between the curves.

Hardness Values for Various Sheet Materials. The following values give tensile and elongation values for brass and hardness values for several materials.

TABLE XVIII

VICKERS PYRAMID NUMERALS

Load: 5 kg.

Grade	Bright Mild Steel		Deep Drawing Mild Steel		Welsh Plate		
	Min.	Max.	Min.	Max.	Quality	Min.	Max.
1. Soft	—	110	—	105	P.C.A.	80	110
2. Medium hard	127	156	-	—	C.R.C.A.	100	120
3. Hard	170	—	—	—	P.C.R.C.A.	95	120

TABLE XIX

TENSILE AND ELONGATION VALUES FOR BRASS OF VARIOUS TEMPERS

	Tons per in.^{2} Minimum	Elongation Percentage on 2 in.
1. Soft	18	46
2. Half hard	24	26
3. Hard	34	8
4. Spring	40	4

TABLE XX

VICKERS PYRAMID NUMERALS

Load: 5 kg.

	Brass		18% Nickel Silver		Phosphor Bronze		Aluminium		Sheet Copper	
	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
1. Soft	—	75	—	100	—	100	—	30	—	70
2. Half hard	95	125	130	165	141	185	30	40	75	120
3. Hard	140	155	170	195	190	228	40	55	125	180
4. Spring	165	—	200	—	235	—	—	—	—	—

Note. The values given in the above tables are due to P. Mabb.

The Jovignot Test. A method of testing sheet metal has been introduced recently by Jovignot in which a clamped circular

plate, the test piece, is subjected to fluid pressure. Rupture occurs eventually as a result of the pressure and deformation. It is found that the sheet deforms to an approximately spherical segment and that the stress at fracture can be calculated by the formula for the strength of a spherical shell, namely

$$\text{stress} = PR/2t$$

where P is the pressure, t the thickness of the sheet and R the radius of curvature $= (r^2 + h^2)2h$. See Fig. 203.

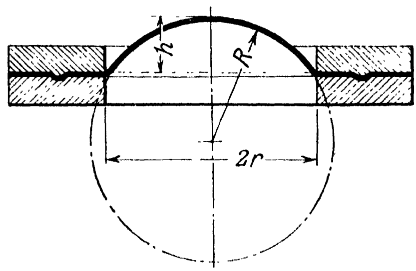


Fig. 203. Jovignot Test

A measure of the ductility can be deduced from the initial and final dimensions of the test piece. The *cupping coefficient*, which is equivalent to the average increase of surface area of the test piece per unit area, is expressed as the ratio h^2/r^2.

The test is now being investigated with a view to its possible standardization.

Rolland and Sorin's Method of Determining Young's Modulus. A novel method of determining Young's modulus has been devised by Rolland and Sorin. The test piece S Fig. 204, of rectangular section, is held firmly in a vertical position by a clamp at its lower end. The upper end carries a platform A on which bear two identical pendulums P and P'. If P be at rest and P' be set in oscillation the slight reactions of the swinging pendulum produce small elastic deformations in the test piece which in turn cause P to oscillate. Eventually the energy of P' is transferred to P, the amplitude of whose oscillations reach a maximum value while the amplitude of the oscillations of P is reduced to zero. Energy is then transferred in the opposite direction until P has been brought to rest and P' again has its largest swing. The cycle is repeated until the energy of the pendulums is finally damped out.

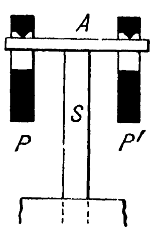


Fig. 204. Pendulum Method of Determining Young's Modulus

The quantity to be observed is the time t which elapses between two successive arrests of the same pendulum. If the

section of the test piece has a breadth b and a depth d in the plane of bending and the length from the point of fixing to the plane of the knife-edges is l, the rigidity K of the bar is the static end deflection for unit load and is given by $K = 3EI/l^3 = Ebd^3/4l^3$ where I is the moment of inertia of the cross-section. The value K determines the amplitude of the movement of the end of the test piece caused by the reaction of the pendulum, and hence the value of t.

If each pendulum has a mass M and time-period T the relation between K and t is

$$K = (2\pi^2M/T^2)(t/T + 2\mu/M)$$

in which μ is the equivalent mass of the test piece. For a test piece of mass m fixed in the manner shown, the value of μ is $33/140\ m$.

As t is generally large compared with T and the pendulums are heavy so that $2\mu/m$ is negligible,

$$K = 2\pi^2Mt/T^3$$

and hence

$$E = 8\pi^2ML^3t/T^3bd^3$$

The chief advantages claimed for the method are that measurements of load and extension are replaced by a single time measurement; the test piece has time to settle down under alternations of loading and give a true modulus; and that the modulus obtained is that at the origin of the stress-strain curve.

The following table compares values of E for several materials obtained by the method described and by the usual tension method using a Martens' extensometer.

TABLE XXI

Comparison of Values of Young's Modulus (E) Determined by the Pendulum Method and by the Direct Method Using a Martens' Extensometer

Metal	E Pendulum Method kg. per mm.^{2}	E Martens' Extensometer kg. per mm.^{2}
Steel	21 000	20 830
Duralumin	7 640	7 460
Aluminium	7 650	7 600
Bronze 1	12 000	11 990
Bronze 2	12 500	12 900

CHAPTER XII

SOME TEST PHENOMENA AND RESULTS

Influence of Form of Test Piece on the Shape of the Stress-strain Diagrams. The shape of the load-extension or stress-strain diagram obtained in a tensile test will vary not only for different metals but also for the same material to an extent depending on the chemical composition; on the mechanical and thermal treatment the material has hitherto received; on the form of the test piece; and on the sensitiveness of the testing machine employed.

The influence of the shape of the test piece is brought out in the diagram Fig. 205. The graphs *a* and *b* represent the load-

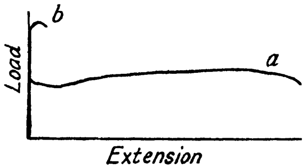


FIG. 205. INFLUENCE OF SHAPE OF TEST PIECE ON THE LOAD EXTENSION DIAGRAM

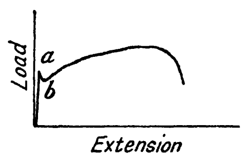


FIG. 206. ILLUSTRATING THE "DROP" AT THE YIELD POINT

extension diagrams obtained with mild steel test pieces 0·6 in. diameter and 3 in. and $\frac{1}{8}$ in. respectively between shoulders.

It will be noticed that the shorter specimen sustained approximately 60 per cent greater load than that carried by the 3 in. specimen and that the drop at the yield point has vanished. The diagrams suggest materials possessing entirely different properties. This example emphasizes the necessity for standardization of test pieces in order to obtain comparable results.

An increase in the carbon content of a steel will likewise cause a marked change in the stress strain-diagram, a high percentage of carbon resulting in little elongation of the test piece and giving a graph similar to *b* of Fig. 205.

The characteristic yield point given by mild steel and wrought iron, at which the specimen elongates without increase of load, or rather, usually with a drop in load, is absent in all other materials. The two points *a* and *b*, Fig. 206, represent

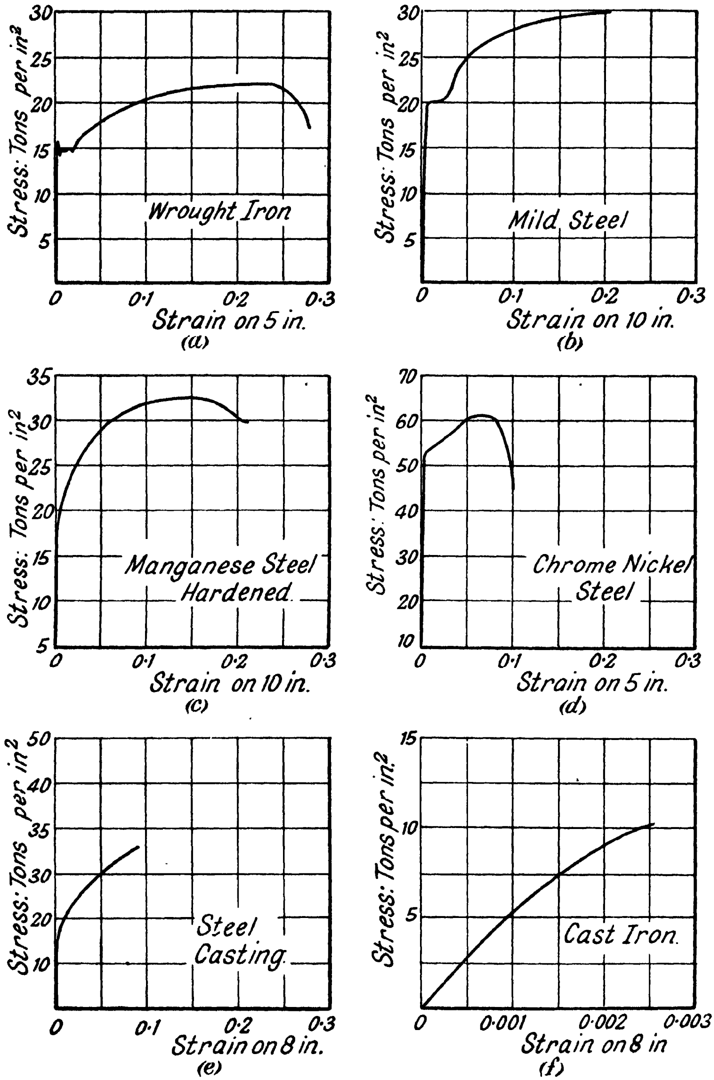


FIG. 207. STRESS-STRAIN DIAGRAM GIVEN BY FERROUS MATERIALS

what are termed the "upper" and "lower" yield points respectively.

The drop in load at the yield point, which is a few per cent in a diagram obtained in a tensile test carried out with the aid of an autographic recorder, has been shown by Dalby, using an optical device (page 133), to amount to 13 per cent, while Cook and Robertson, who employed a special arrangement to overcome the inertia of the elastic system, obtained a reduction of 27 per cent.

The upper yield point is affected by the speed of testing.

Stress-strain Diagrams for Metals in Tension. Various tensile stress-strain diagrams are shown in Fig. 207. (*a* to *f*). The forms (*a*) and (*b*) given by wrought iron and mild steel respectively are very similar. Whether or not a drop is indicated at the yield point will depend on the sensitiveness of the machine and the recording apparatus.

Diagram (*c*) is a curve for hard manganese steel. In the annealed state of the material the stress-strain curve is similar to (*b*) for mild steel. High tensile alloy steels show less and less extension as the tensile strength rises. A curve obtained by Dalby with a 60 ton chrome nickel steel is shown in (*d*). Curve (*e*) is from a steel casting in the "as received" condition. Cast iron (*f*) gives a curve no part of which is straight. Young's modulus is then usually determined as the tangent modulus at the origin.

Stress-strain diagrams for several non-ferrous metals and alloys are given in Fig. 208 (*a* to *f*). Copper shows no definite yield point. A yield point must therefore be determined in an arbitrary manner. It is customary to draw a line parallel to the tangent to the curve at the origin at a point on the strain axis representing a strain of 0·15 or 0·2 per cent. The point of intersection of the line so drawn with the curve is taken to be the yield point.

The curve for aluminium bronze, Fig. 208 (*c*), shows an abrupt change at the yield point.

Tin and zinc show a striking difference from other metals in that the yield point and the point of maximum load are almost coincident, the curve falling away rapidly as the test proceeds. The forms of the curves given by tin and zinc persist in some alloys in which these constituents are present, notably gun metal and phosphor bronze, especially if the tin predominates. Brass, however, gives a rising curve (Fig. 208 (*b*)).

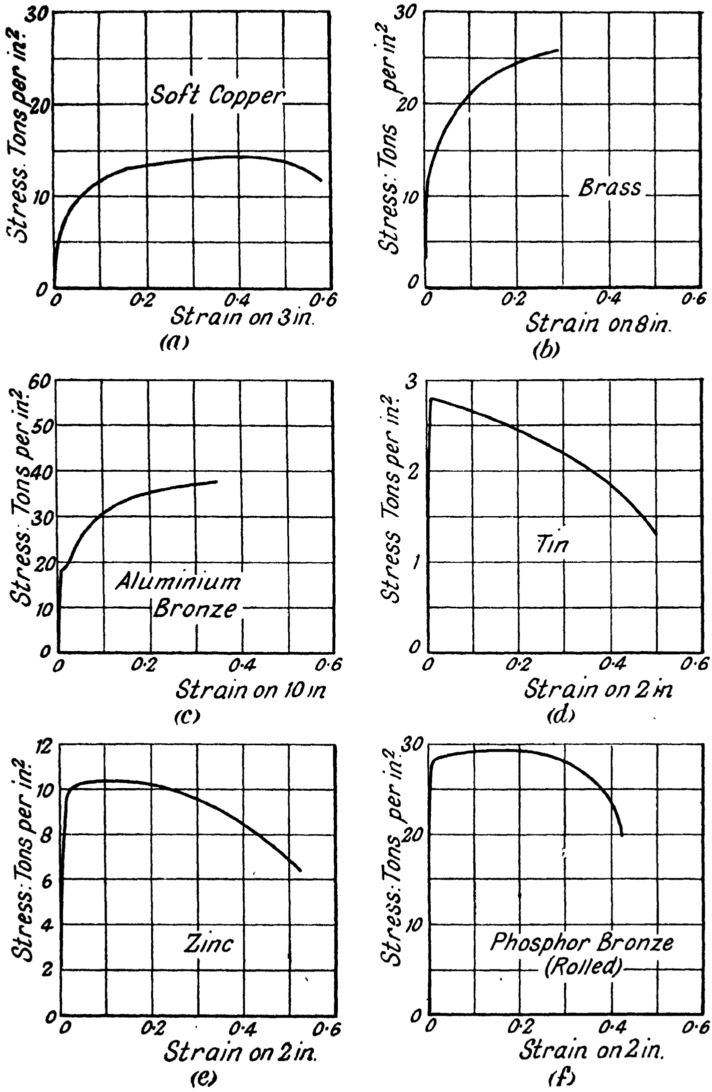


FIG. 208. STRESS-STRAIN DIAGRAMS GIVEN BY NON-FERROUS MATERIALS

Effects of Overstrain. If a test piece of mild steel be loaded in tension to a point in excess of the yield point, *A*, Fig. 209, and then unloaded, the graph connecting load and extension will follow an approximately straight line *AB*, and if the test piece be immediately reloaded it will be found that the limit of proportionality has been lowered while the yield point has been raised to a point such as *C*. Further loading will yield the curve *CD*, coinciding with a continuation of the original curve. If several days are allowed to elapse before reloading, the yield point will be found to be still higher as indicated at *E*. The time that elapses between the loading and unloading has considerable influence on the shape of the resulting stress strain curve. In some tests by Ewing it was found that if reloading followed

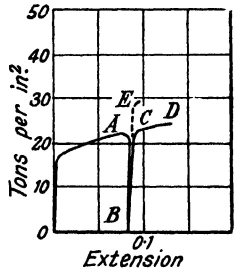


FIG. 209. EFFECT OF LOADING AND UNLOADING

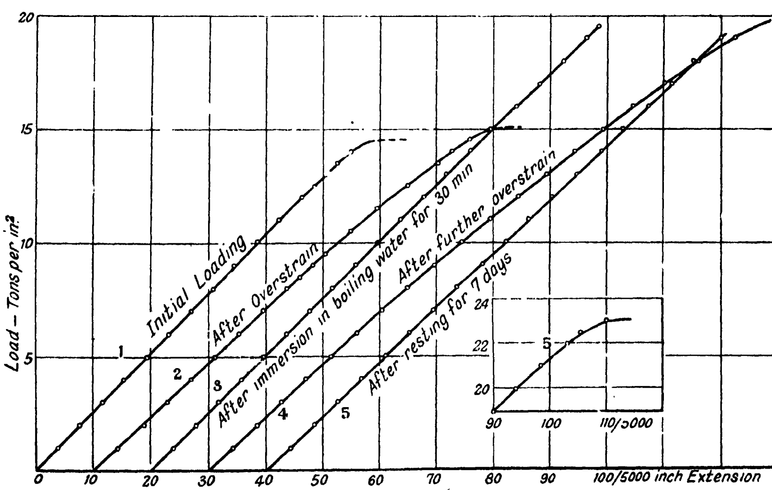


FIG. 210. EFFECT OF REST AND MILD HEAT TREATMENT ON MILD STEEL

in ten minutes after overstrain beyond the yield point, the material failed to obey Hooke's law. After five days the

specimen partially recovered its elasticity and after twenty-one days had completely recovered it.

Mild heat treatment after unloading, say by boiling for twenty minutes at 100° C., furthers the recovery of the elastic properties in a remarkable degree. This was first shown by Muir in 1899. It is thus seen that the general effect of overstrain is to raise the yield point and to reduce the limit of proportionality.

The results of some tests by the Author on a specimen of mild steel illustrate this point (Fig. 210).

Low carbon steels generally recover their elastic properties after a rest or after boiling, but alloy steels are lacking in this respect. The graph in Fig. 211 is from a test by Dalby on a

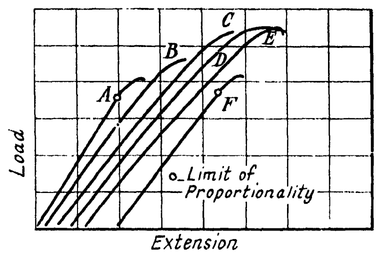


FIG. 211. RESULT OF TEST ON NICKEL STEEL

0·3 carbon, 3·68 nickel steel. Under the initial test, curve A, the limit of proportionality was 58 000 lb. per in.^{2} and the yield point 67 000 lb. per in.^{2} The material was stretched 2 per cent on a 5 in. gauge length and the second curve B shows that the limit of proportionality has vanished. Curve C was obtained after 6 per cent stretch, and curve D after a lapse of 24 hours, the bar having been turned down to a slightly smaller diameter. After boiling for one hour curve E was obtained, no recovery in elasticity being apparent. Finally the specimen was heated to 550° C. and kept at this temperature for 30 min. This treatment restored the elasticity. Both the limit of proportionality and the yield point were raised slightly as shown by the curve F.

Phenomenon of Strain Hardening. The phenomenon of hardening due to plastic distortion is termed *strain hardening*. The effect on a metal is to make it harder and less ductile. Such hardening disappears on annealing. A material which has been cold worked will exhibit tensile properties similar to

those of the overstrained material just discussed. Improvement in the tensile properties of a material in the direction of loading is not necessarily accompanied by improvement in its compression properties in the same direction.

If a single crystal of metal be tested in tension its strength will depend on the direction of the pull relative to the axes of the crystal. Plastic deformation of such specimens involves sliding or slipping along certain planes, and the commencement of slip depends on the shear stress along the plane and is independent of the normal stress. Increase of load increases the shear stress and the number of planes on which sliding occurs, with consequent further elongation of the specimen. The increase in stress necessary to continue the stretching represents the strain hardening of the crystal.

In a brittle material, fracture occurs due to the overcoming of cohesion on a certain crystallographic plane by the normal tensile stress. A commercial metal, on the other hand, exhibits the average effect on all the crystals of which it is composed. The result is that its mechanical properties are very nearly independent of direction.

Microscopic observation of a polished specimen under a tensile test shows a number of lines on the surface known as slip bands, due to the slipping of individual crystals under the action of the stress. The sliding stops at the crystal boundary. Some crystals may be less favourably situated than others to withstand tensile stress, and this is believed to be the cause of small deviations from Hooke's law in materials which are generally assumed to be elastic.

In ductile materials, especially those with a well defined yield point, considerable sliding along the planes of maximum shear takes place when the yield point is reached. These planes are indicated by the Lueder's lines which appear on the surface of a polished specimen an at angle of about 45° to the axis of the specimen. Individual crystals become strain hardened by distortion and on reloading the specimen the yield point is found to be higher than before.

Characteristic of Tensile Fracture in Ductile Materials. The strength of a test piece may be regarded as due to its resistance to sliding or to its resistance to separation. The associated types of failure that occur are termed *sliding failure* and *separation failure,* the former occurring in ductile and the latter in brittle materials. The relation between the two types of resistance

varies throughout a test. Resistance to sliding seems to increase as the velocity of deformation increases and decreases with rise of temperature. The resistance to separation is not affected to the same degree.

The type of fracture shown by a material depends on the conditions under which fracture takes place, as for instance whether or not plastic deformation due to sliding is prevented. The load extension diagram of a ductile material when flow is prevented by a groove in the test piece is similar to the curve (*b*), Fig. 205.

In a three-dimensional stress system such as that represented in Fig. 22, Chapter I, the maximum shearing stress is $(p_1 - p_2)/2$,

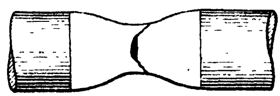


FIG. 212. LONGITUDINAL SECTION THROUGH FRACTURE IN A DUCTILE MATERIAL

and if p_1 and p_2 be very nearly equal the maximum tensile stress may be many times the shear stress. Such conditions in a ductile material are productive of a brittle fracture.

In a specimen of mild steel under tension a three-dimensional stress condition obtains in the middle of the specimen. The metal at the neck is subject to tension in the radial as well as in the axial direction, and fracture takes place first by the formation of a crack in the centre of the section and then by yielding and sliding along planes inclined at 45° near the boundary. The result is the well-known cup-shaped fracture that appears when a bar of mild steel is tested to destruction in tension. Fig. 212 shows the appearance of the longitudinal section through a fractured test piece, the broken parts having been first fitted together and the metal in the neighbourhood of the break filed away.

On account of the deformation only the inclined portions at the boundary make contact, leaving a hollow space in the centre of the break. The diagram given by Professor Haigh to represent the state of affairs is shown in Fig. 213. On the little cube of material at the centre there is no tendency to make the metal shear, while the inner part of the neck has to pull in all directions in order to keep the curved outer fibres in equilibrium.

Hysteresis. Experiment shows that the elongation of a test-piece does not immediately follow the application of the load and that the specimen continues to elongate when the load has reached its final value. Tests made on single crystals show that if the specimen be loaded quickly according to the load extension line *oa*, Fig. 214, a lowering of temperature will occur due to increase in volume. If the application of the load be sufficiently rapid the process may be regarded as adiabatic, that is, there is no exchange of heat between the bar and the surrounding medium. The bar, however, gradually warms up and a slight additional elongation takes place represented by *ab*, the load remaining constant throughout this change. Rapid unloading gives the line *bc*, a rise in temperature occurring because of the decrease in volume. On cooling, the bar contracts by the amount *co* and resumes its original state. The line *oa* gives the adiabatic modulus of elasticity while the line *ob* would give the isothermal or constant temperature modulus.

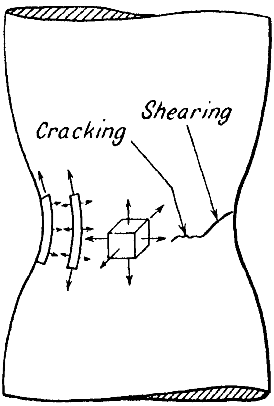


FIG. 213. FRACTURE OF A DUCTILE MATERIAL (*Haigh*)

In actual materials this time effect is much greater than in the single crystal and cannot be explained by purely thermo-dynamic considerations. One explanation is that it is due to the continued sliding within unfavourably situated crystals. The time effect after unloading is explained as due to the residual stresses which continue to produce sliding in crystals unfavourably orientated, thus causing creep in the material after the load is removed.

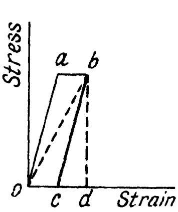


FIG. 214. ELASTIC HYSTERESIS

Many years ago Lord Kelvin gave the ratios of the quick to the slow Young's moduli for several materials, among others—

Iron . .	1·0026
Copper .	1·00325
Tin . .	1·0036
Zinc . .	1·008

The area enclosed by the loop in the diagram, Fig. 214, represents the dissipation of a certain amount of energy during the cycle of operations. With sufficiently delicate methods of measurement it is found that similar phenomena occur with metals generally, the strain lagging behind the stress even with stresses much below the value usually taken as representing the elastic limit of the metal. If a test-piece be loaded and unloaded repeatedly with tensile and compressive loads of the same final magnitude the resulting stress-strain diagram will take the form of a loop, Fig. 215. The shape of the loop formed

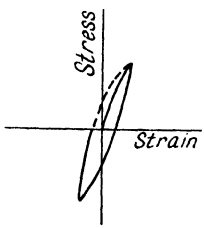


FIG. 215. ELASTIC HYSTERESIS LOOP

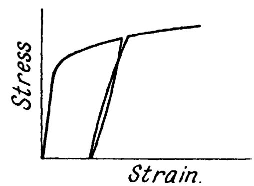


FIG. 216. HYSTERESIS LOOP OBTAINED WHEN LOADING AND UNLOADING BEYOND THE ELASTIC LIMIT

after the initial loading and unloading will depend on the history of the specimen. The size of the loop, termed the *hysteresis loop*, will depend on the material and on the range of stress. This is *elastic hysteresis*.

If a metal be stretched beyond the elastic limit the recovery of elasticity on unloading is usually incomplete, and on reloading, the loading line forms a loop with the previous unloading line. This hysteresis loop is much larger than the hysteresis loop within the elastic limit (Fig. 216). The phenomenon, so far, has not been completely explained.

Stress-strain Curves for Metals in Compression. Stress-strain curves for metals in compression exhibit as much divergence as do those for metals in tension.

The compression curve for cast iron, Fig. 217 (*a*), is similar in form to the curve for tension. Wrought iron, Fig. 217 (*b*), and mild steel, Fig. 217 (*c*), show a yield point, but less well defined than in tension and from this point onwards the curve rises continually. In this region the stress plotted is the nominal

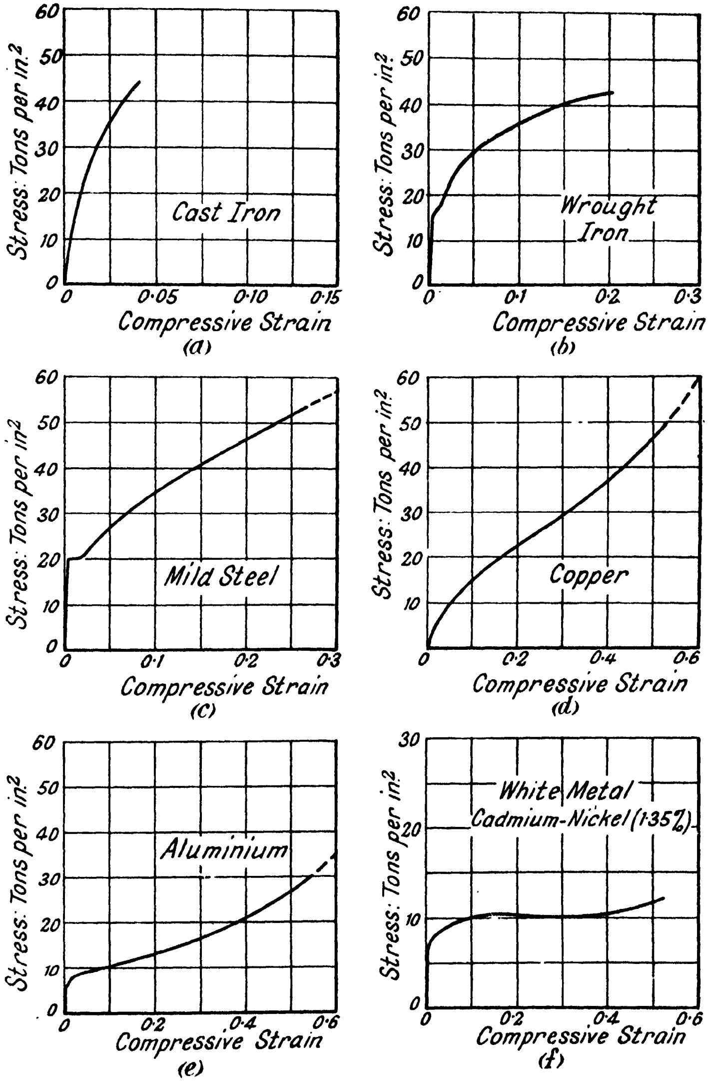


FIG. 217. STRESS-STRAIN CURVES OBTAINED IN COMPRESSION TESTS

Influence of Temperature on Mechanical Properties of Metals. The mechanical properties of metals are severely modified by

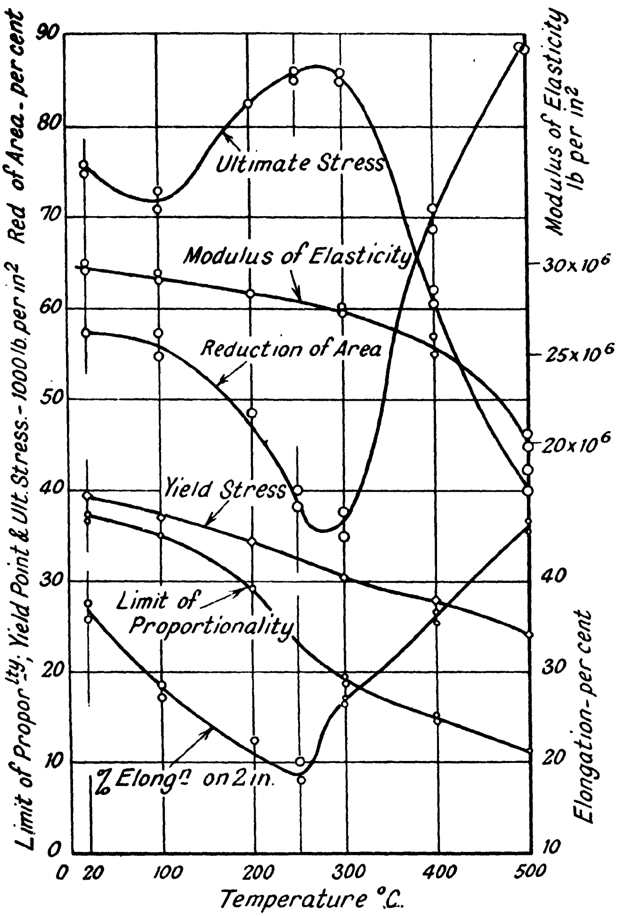


FIG. 220. TENSILE TESTS AT HIGH TEMPERATURE ON MEDIUM CARBON STEEL

(0·37 C., 0·63 Mn., 0·11 Si.)

(*Westinghouse Electrical and Manufacturing Co., Pittsburgh*)

rise of temperature when this exceeds about 200° C. The tensile strength of mild steel increases by some 30 per cent up to about 300° C. and then falls considerably as the higher temperatures

are reached. At 500° C. the strength is only about one-half the strength at atmospheric temperatures. On the other hand the strength rises as the temperature is reduced beyond zero.

The elastic limit falls continuously as the temperature rises,

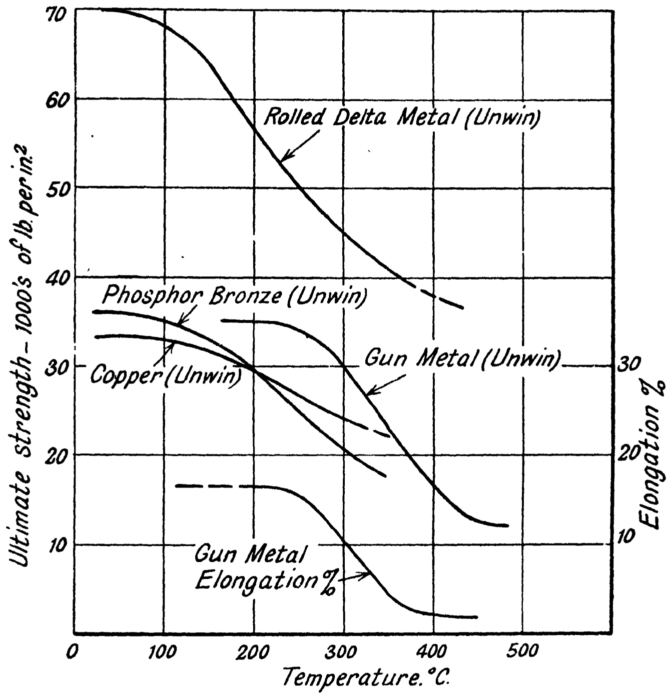


FIG. 221. INFLUENCE OF TEMPERATURE ON PROPERTIES OF NON-FERROUS MATERIALS

while the elongation diminishes up to about 200° C. and then increases.

Different varieties of steel exhibit widely different temperature characteristics, but a general idea of the variation in properties with temperature can be gleaned from Fig. 220, which shows the effect of temperature on the properties of a medium carbon steel. Curves showing the variation in tensile strength with temperature for various non-ferrous alloys are given in Fig. 221.

The strength of cast iron increases up to about 500° C. and then diminishes.

The results of some torsion tests on mild steel are given in Fig. 222.

Impact tests of steel show a decrease in the Izod or Charpy value up to 500° C., after which the values increase. Fig. 223

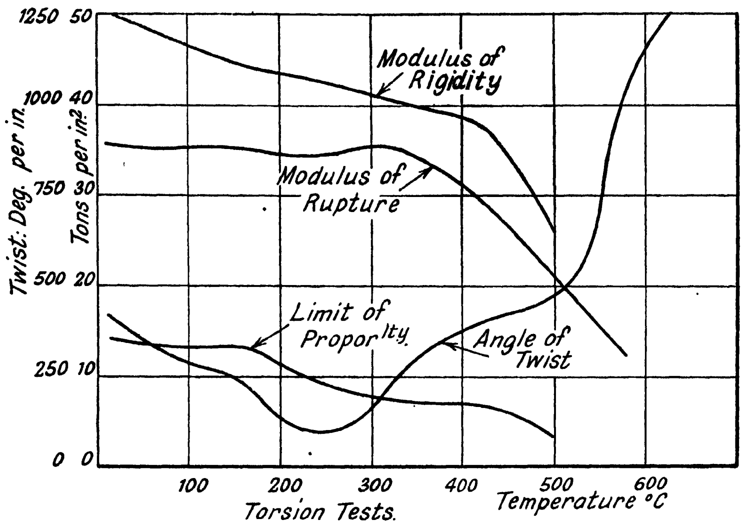


FIG. 222. RESULTS OF TORSION TESTS OF MILD STEEL AT HIGH TEMPERATURES

shows the results of tests on medium carbon steel (curve A) and on mild steel (curve B).

Hardness tests show irregularities over a temperature range of 0° C. to 700° C., the hardness numbers rising and falling throughout this range.

Fatigue strengths appear to reach a maximum in the neighbourhood of 250° C., but some steels, such as nickel-chrome, do not exhibit this increase.

Reference should be made to the Reports mentioned at the end of the book, where detailed information relating to the above tests will be found.

The study of the behaviour of metals at high temperatures has become of prime importance owing largely to developments

in high pressure steam plants, and intensive research is now being pursued.

At high temperatures the duration of the test has a marked

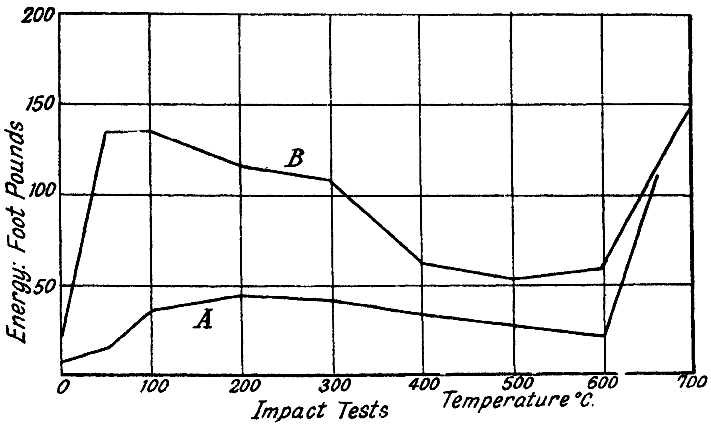


FIG. 223. IMPACT TESTS ON STEEL AT HIGH TEMPERATURES

influence on the results obtained, the load needed to produce fracture becoming smaller as the duration of the test is increased.

On the experimental side specimens are submitted to a constant load and temperature and the progressive creep under these conditions is investigated. If the results are plotted as an extension-time diagram a curve similar to that in Fig. 224 is obtained. The portion OA represents the initial extension. The rate of extension increases rapidly at first, but after the state represented by point A is reached it remains practically constant over a range AB. After the state corresponding with B is reached the rate of extension increases and fracture ultimately occurs. The life of the test piece lies within the range AB. If the stress be reduced the slope of AB decreases, but there appears to be no limiting creep stress at which the test piece can resist stress and high temperature indefinitely.

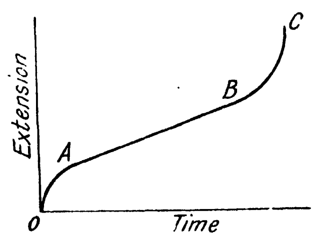


FIG. 224. EXTENSION-TIME DIAGRAM

Two effects come into play in such tests—

(1) Hardening of the metal due to plastic strain.

(2) Softening of the metal due to the prolonged action of high temperature.

In the extension-time curve, Fig. 224, the reduction in the rate of extension over the portion *OA* of the curve is due to strain hardening. The constant rate of extension over the range *AB* is brought about by the removal of the strain hardening produced by creep by the softening effect of the high temperature.

FIG. 225. METROVICK SLOW TENSILE TESTING EQUIPMENT
(*Metropolitan-Vickers Co. Ltd.*)

The Metrovick Slow Tensile Testing Equipment. The Metropolitan-Vickers Company have developed both slow tensile testing and creep testing equipment. The slow tensile testing machine is shown in Fig. 225. The load is applied through a train of gearing which provides rates of strain of 0·001 and 0·25 strain per minute as desired.

The specimen is connected with the loading mechanism through a short length of steel rod and a special chuck coupling which is visible in the illustration above the lower cross member of the machine. This simplifies the disconnection of the motor-driven loading mechanism and the connection of the loading gear that is fitted when it is desired to use the machine for creep testing. Load and extension are both recorded automatically by a photographic method, but direct visual measurement may be made. The temperature of the specimen can be finely adjusted up to

a maximum of 900° C. and is controlled automatically to within very close limits with a high degree of uniformity throughout the length of the specimen.

The slow tensile testing equipment serves a useful function in that it enables a range of steels to be placed in relative order of their creep resistance. Research has shown that for constant rates of strain the ultimate tensile strength will vary considerably with the temperature, and conversely, with constant temperature it will vary with the rate of strain. There is, in fact, a particular rate of strain which gives the optimum relation between the thermal hardening and the rate of increase of load and results in a maximum tensile strength for that temperature.

Generally, steel parts subjected to stress at elevated temperatures will operate for long periods; and thermal hardening and softening phenomena will not play any important part in their behaviour in service as these phenomena will take place during a short initial load period which (with the extremely slow rates of straining common under practical conditions) is negligible in relation to the time the part is in service.

Tensile tests, therefore, which are intended to indicate the relative creep resistance of materials under service conditions should be carried out in such a way as to ensure freedom from serious interference by thermal hardening effects, this requirement being met by testing at a suitably high temperature. The standard test temperature adopted by the Metropolitan-Vickers Company is 550° C. where the material represented by the specimen is to be used up to 500° C. When the service temperature is above 500° C. special precautions have to be taken against scaling of the specimen, and a test temperature of 550° C. or the service temperature, whichever is the higher, is employed.

It has been found that the slower rates of straining provide a better discrimination between steels, and the selection of a standard rate of straining resolves itself into the choice of a rate that will permit of a test being completed in a working day.

The results of some tensile tests on steels under various rates of straining are given in Fig. 226. A suitable rate of testing is 0·001 strain per minute.

The autographic recording apparatus consists of a drum inside the cylinder shown on the top of the machine in Fig. 225, and which is rotated through a vertical shaft driven from the

loading mechanism. The rotation is proportional to the extension of the test piece.

In one side of the otherwise light-proof cylinder surrounding the drum is a vertical slit, along which is fitted a glass tube of square section containing mercury and connected at the bottom to a metal reservoir. The base of the reservoir consists of a

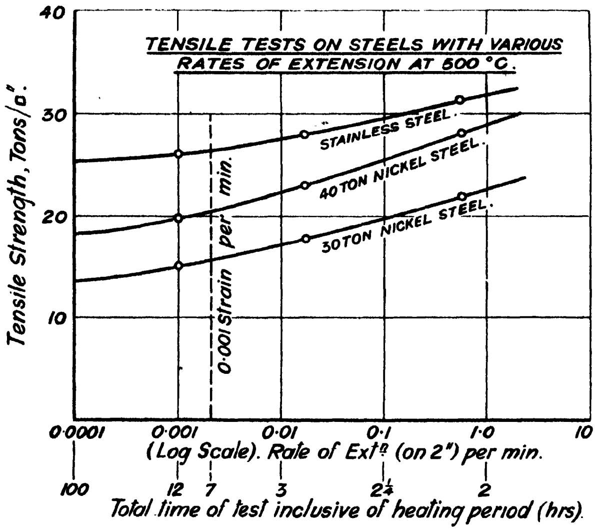


FIG. 226. TENSILE TESTS ON STEELS WITH VARIOUS RATES OF EXTENSION AT 500° C.
(*Metropolitan-Vickers Co. Ltd.*)

heavy diaphragm connected to the top end of the specimen. The deflection of the diaphragm as the load comes on the specimen varies the height of the mercury column in the tube.

A beam of light is projected through the slit in the cylinder on to the sensitized paper fitted to the recording drum, the mercury column acting as a shutter to vary the depth of the exposed portion of the record in accordance with the load on the test piece. A typical stress-strain diagram obtained with this apparatus is shown in Fig. 227.

The maximum stress exerted with the standard machine is 45 tons per in.^{2} on 0·1 in.^{2} of section. Specimens up to 8 in. gauge length can be accommodated.

Creep Tests. With regard to creep tests it may be remarked that tests of which the results have been published involve a duration which is not comparable with that of the service life of the material used in steam plants. Hence, in this respect, even tests of the longest duration can only be regarded as short-time tests.

Further, the majority of such tests have been carried out at stresses higher than working stresses, since most creep testing

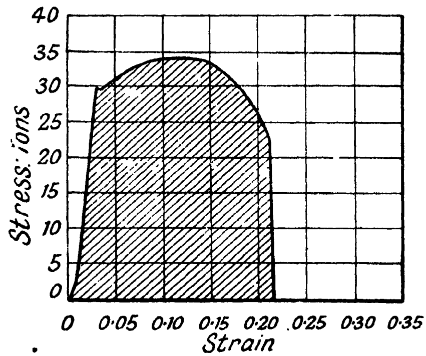


FIG. 227. AUTOGRAPHIC STRESS-STRAIN DIAGRAM MADE BY THE METROVICK PHOTOGRAPHIC RECORDING APPARATUS
(*Metropolitan-Vickers Co. Ltd.*)

machines are not capable of measuring in a reasonable time the slow rates of creep permissible under service conditions.

Practical requirements necessitate the measurement of rates of strain of 10^{-8} per hour within a reasonable time. The essentials for such apparatus are difficult of achievement. They are—

(*a*) Means for adjustment and measurement of the actual temperature of the specimen within 1° or 2° C.

(*b*) Means for obtaining constancy of temperature over long periods to within ½° C. at all temperatures up to 850° C.

(*c*) Means for obtaining uniform temperature along the specimen within 1° or 2° C.

(*d*) Accurate and constant loading arrangements.

(*e*) Extensometer equipment capable of reading strains of 10^{-6} directly.

Metrovick Single-Unit Creep Testing Equipment. The most recent design of Metrovick single-unit creep testing equipment operating on a.c. supply is shown in Fig. 228. It is capable of

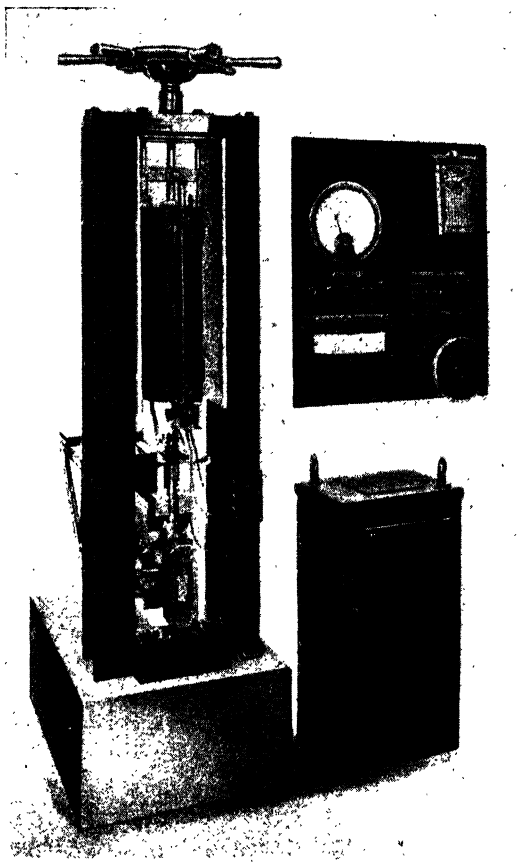


FIG. 228. METROVICK SINGLE UNIT CREEP TESTING EQUIPMENT
(*Metropolitan-Vickers Co. Ltd.*)

continuous service at temperatures up to a maximum of 700° C. The specimen temperature is thermostatically controlled to within 1° C. The design provides for a stress range of 0·5 to 20 tons per in.² on a test piece having a 5 in. gauge length

and a cross-sectional area of 0·1 in.^{2} When using the dial indicator which is fitted to the end of the loading lever creep rates down to 10^{-5} strain per hour can be measured. For greater accuracy mirror extensometers must be used.

Provision is made for lowering the load on to the test piece without shock at the commencement of a test. After switching on the current, the furnace heats up to the selected temperature at which the thermostat operates and controls the temperature to within the limits stated.

The equipment includes apparatus for the determination of proportional limits at elevated temperatures.

The furnace and the extensometer introduced by the Metropolitan-Vickers Company both differ from the conventional types. In the construction of the furnace a steel tube is used in place of the usual silica tube, and the extensometer is secured to the specimen by means of split clamps screwed together over specially enlarged portions at either end of the gauge length. For long-time high temperature tests experience has demonstrated the superiority of this method over the usual pointed-screw or knife-edge method of fixing. The instrument is shown in Fig. 229. Two mirrors with telescopes and scales are employed and the strain on an 8 in. gauge length may be read to $0{\cdot}25 \times 10^{-7}$.

FIG. 229
EXTENSOMETER FOR HIGH TEMPERATURE MEASUREMENTS
(*Metropolitan-Vickers Co. Ltd.*)

Making allowance for some slight stagger of the plotted points it has been found in practice that constant creep rates of 10^{-8} strain per hour can be determined with reasonable certainty in from 400 to 500 hours. Fig. 230 shows a typical curve.

The Damping Capacity of Metals. The phenomenon termed "elastic hysteresis" has been considered previously and its connection with the fatigue limit discussed. Within the last decade the phenomenon has assumed additional importance, largely by virtue of its effect in vibrating elastic systems when under a condition of resonance, as, for example, when a crankshaft passes through a critical speed. Such

oscillations are limited in amplitude by the damping forces associated with the system, one of which is operative through the internal friction of the material. When the system is left

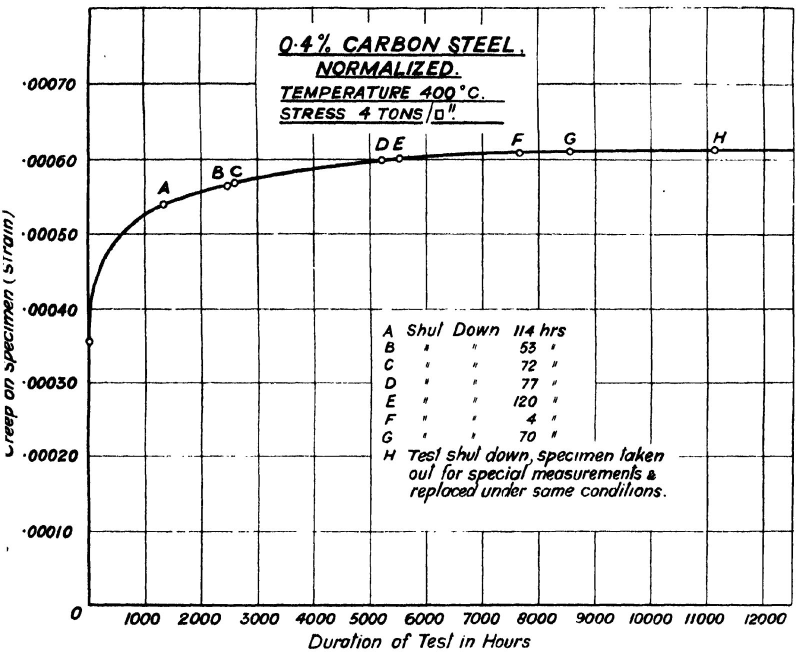


FIG. 230. TYPICAL CURVE SHOWING NATURE OF TESTING PERFORMED WITH METROVICK CREEP TESTING EQUIPMENT

0·4 per cent carbon steel normalized. Temperature 400° C., stress 4 tons/in.[2]

(Metropolitan-Vickers Co. Ltd.)

to itself after having been set in oscillation, it is brought again to equilibrium by the dissipation of its vibrational energy and the measure in which mechanical hysteresis assists in the restoration would be manifest in the area of the hysteresis loop if this were obtained. From the present viewpoint this property is referred to as the "damping capacity."

Damping capacity is defined as the amount of work dissipated into heat by a unit volume of the material during a completely

reversed cycle of unit stress. It is usually measured in inch-lb. per cubic inch per cycle.*

The area of the hysteresis loop $ABCDA$ (Fig. 231) represents the energy dissipated throughout the cycle, and is a direct measure of the damping capacity. The ordinate at A represents the maximum stress and the energy expended in reaching this state is represented by the area under the line OA.

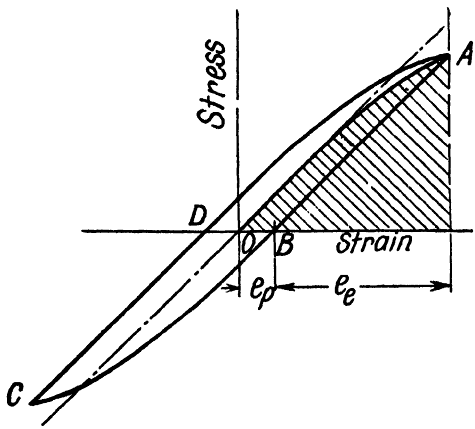


FIG. 231. HYSTERESIS LOOP FOR REVERSED STRESS CYCLE

Of the total strain at A the part denoted by e_e is elastic strain and that denoted by e_p is plastic strain.

In determining the hysteresis loop the ordinary testing machine is too rough an instrument for the purpose, and it becomes necessary to resort to loading by dead weights and to use a very sensitive extensometer. Since loading by dead weights is an impracticable procedure with tension specimens the determination is usually carried out by torsion or bending, as the leverage due to the length of beam or length of torque arm, as the case may be, permits of high stresses being realized in the test piece with comparatively small weights. This method of procedure is the *Static Method*.

Several other methods are available—

The Starting Method employs a torsional fatigue-testing machine. From the commencement of the test the damping

* 14·2 cm.-kg. per c.c. per cycle

energy is continuously converted into heat which raises the temperature of the test piece. If this temperature be plotted as a function of the time, the slope of the resulting graph, which depends on the maximum fibre stress, can be shown to be proportional to the damping capacity. A series of such tests enables the variation of damping capacity with maximum stress to be ascertained.

The specimen used needs to be of fair size and obvious precautions must be taken to secure correct measurements of the temperature.

The Equilibrium Temperature Test employs a similar apparatus but here the test is carried on until the temperature of the specimen is practically stationary. This final temperature is proportional to the damping capacity, but it is necessary in this case to determine also the factor of proportionality.

The Energy Input Method. In this, a somewhat rough method, the damping is obtained by measuring the energy input in the manner described on page 223.

All the foregoing methods require elaborate and expensive apparatus.

The Free Vibration Method. For the direct determination of damping capacity the method of free vibrations is the simplest and most convenient.

The specimen A, Fig. 232, is clamped in position in a heavy frame by means of the bolts B_1. To the centre of the specimen is clamped an inertia bar N. The distance piece H is placed between the point of the screw J and the ball K, the screw being adjusted to hold the distance piece in position without moving the arm M which is an extension of the inertia bar. Torsion is applied to the specimen by rotating the screw J, thus causing a twist of the specimen. Attached to the inertia bar is a stylus E which makes contact with a celluloid disc D driven by a clockwork mechanism.

The frame is suspended by a cord with a hook passed through the ring C. The clock is started and the distance piece H sharply pulled out, thus leaving the inertia bar vibrating freely. The consequent oscillations are recorded on the celluloid disc and the final record is examined under a microscope. The suspension of the apparatus by means of a cord during a test is necessary to prevent leakage of energy from the system.

The instrument illustrated is that made by the Cambridge

Instrument Co., Ltd., who have perfected the stylus-on-celluloid method of recording. The important feature is the specially formed point of the stylus which under very slight pressure leaves a well-defined line, and the record is practically

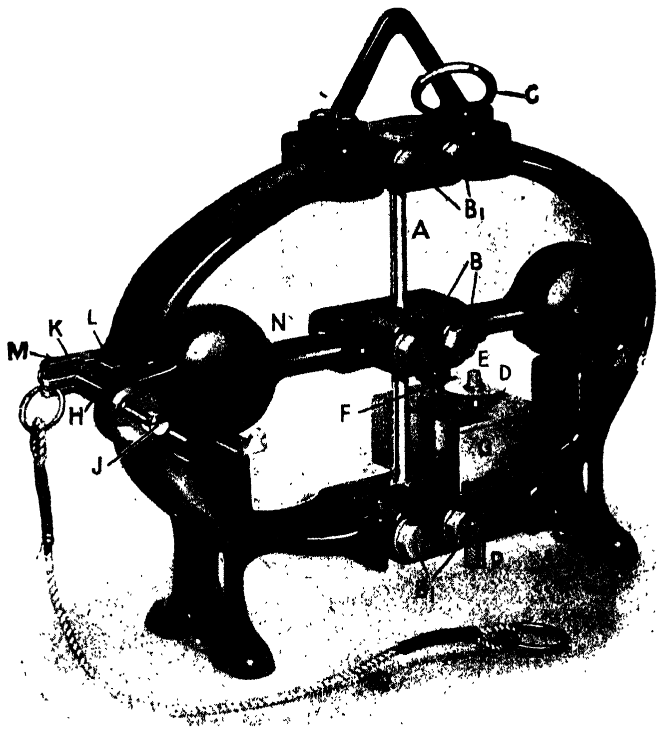


FIG. 232. THE CAMBRIDGE TORSIONAL DAMPING RECORDER

unaffected by friction. Anything in the nature of a needle point is quite unsuitable since apart from the jagged line produced by the point, the accompanying friction seriously impairs the accuracy of the record. When, however, metals of very low damping capacity are being tested an optical method of recording becomes imperative.

The resulting record is that of a damped oscillation (Fig. 233), and from it the "specific damping capacity" (p) can be readily obtained.

Specific damping capacity is defined as the ratio loss of energy per cycle/maximum energy of the cycle.

The logarithmic decrement of the oscillation is the natural logarithm of the ratio of any two successive amplitudes when

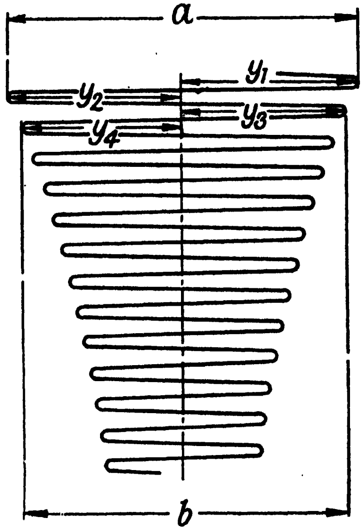


FIG. 233. METHOD OF CALCULATING THE DAMPING RATIO

the amplitudes decrease exponentially, both taken in the same sense. Hence for any amplitude y, the logarithmic decrement is

$$\log_e \frac{y + \Delta y}{y}$$

where Δy represents the decrease of amplitude per cycle. But

$$\log_e \left(\frac{y + \Delta y}{y} \right) = \log_e \left(1 + \frac{\Delta y}{y} \right)$$
$$= \frac{\Delta y}{y} - \tfrac{1}{2} \left(\frac{\Delta y}{y} \right)^2 + \tfrac{1}{3} \left(\frac{\Delta y}{y} \right)^3 - \ . \ . \ .$$

or neglecting terms of the second and higher orders,

$$\delta = \frac{\Delta y}{y}$$

Since the elastic energy required to twist the bar is proportional

to the square of the amplitude, the vibrational energy w at the limit of swing is ky^2, and one cycle later is

$$w - \Delta w = k(y - \Delta y)^2$$

where w is the amount of energy dissipated and Δy is the decrease of amplitude. Hence

$$\Delta w = 2ky\Delta y,$$

whence the specific damping capacity is given by

$$p = \frac{\Delta w}{w} = \frac{2ky\Delta y}{y^2} = \frac{2\Delta y}{y} = 2\delta$$

or, the specific damping capacity is twice the logarithmic decrement.

Measurements may be expressed in terms of the damping ratio R, which is the ratio of the difference between the energy values at the limits of successive amplitudes of opposite sign to the maximum energy. Thus

$$R = \frac{y_1^2 - y_2^2}{y_1^2}$$

Here $y_2 = \lambda y_1$, $y_3 = \lambda y_2$, $y_4 = \lambda y_3$, assuming constancy of λ, and $R = 1 - \lambda^2$.

Referring to the figure, let

$$a = y_1 + y_2 = y_1 (1 + \lambda)$$
$$b = y_3 + y_4 = y_1\lambda^2 (1 + \lambda)$$

then, $$a - b = y_1 (1 + \lambda) (1 - \lambda^2)$$

and $$\frac{a - b}{a} = 1 - \lambda^2 = \delta,$$

therefore $$R = \frac{a - b}{a}$$

and the percentage damping ratio is given by

$$100 \left(\frac{a - b}{a}\right)$$

Since $$p = \frac{y_1^2 - y_3^2}{y_1^2} = 1 - \lambda^4$$

we have in terms of a and b,

$$\begin{aligned} p &= (1 - \lambda^2)(1 + \lambda^2) \\ &= R(1 + 1 - R) \\ &= \frac{a - b}{a}\left(2 - \frac{a - b}{a}\right) \\ &= \frac{a^2 - b^2}{a^2} \end{aligned}$$

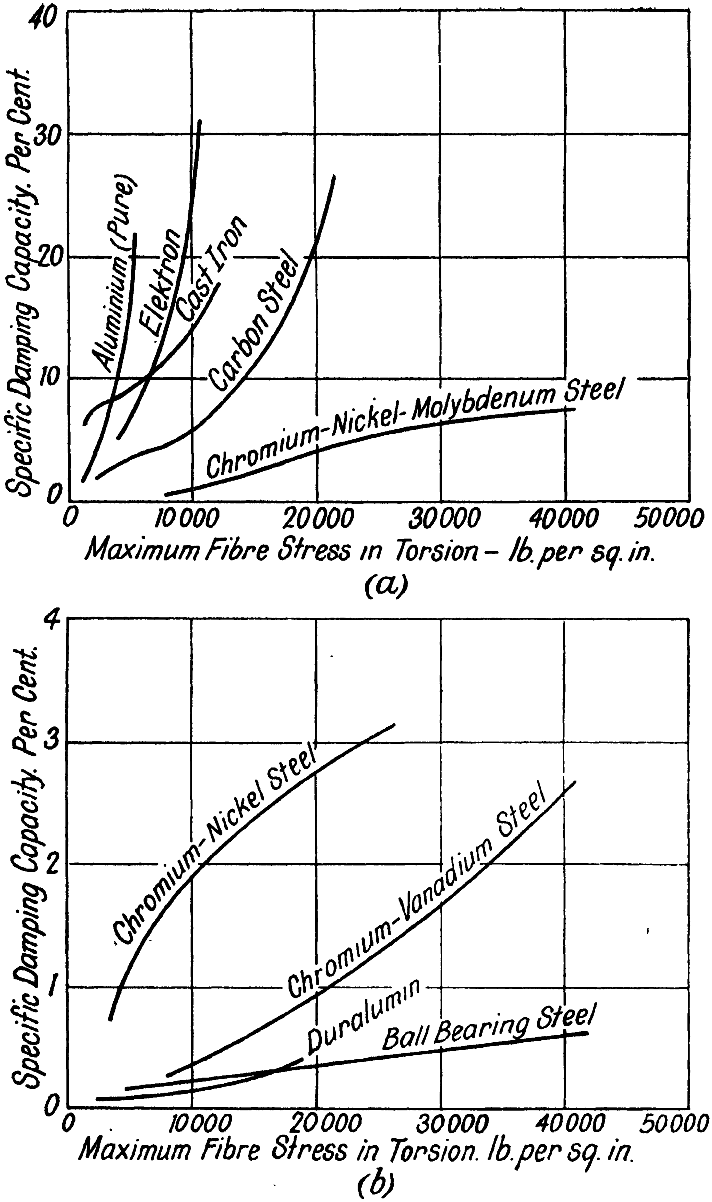


FIG. 234. DAMPING CHARACTERISTICS OF VARIOUS MATERIALS (HEYDEKAMPF)

The damping capacity of a metal depends on its previous history and does not become fully stabilized until the material has undergone a considerable number of oscillations. In practical testing it is therefore customary to subject the test piece to a certain amount of pre-stressing in a torsional fatigue testing machine.

For a given material the damping capacity depends on the amplitude of the stress and follows a law of the form

$$W = AS^x$$

where S = the stress amplitude
A = the hysteresis constant
x = the hysteresis exponent
W = the energy in inch-lb. per cubic inch per cycle.

The values of the parameters A and x vary widely for different steels, and also vary for different ranges of tress. The frequency of stress alternation appears to have little effect provided the temperature of the test piece remains constant.

Curves showing the variation of specific damping capacity with maximum torsional fibre stress for several materials of high and low damping capacities are given in Fig. 234.

It will be observed that while cast iron, aluminium, and electron and carbon steel have high damping capacities, the chrome-nickel steels and duralumin show low damping capacity.

A material of high damping capacity is superior to a material of low damping capacity. The severe local concentrations of stress which occur at flaws and surface scratches are liable to start fatigue cracks. Materials differ in their degree of sensitivity towards these effects and the manner in which a material behaves in this respect is termed "notch-sensitivity."

If a material can endure an indefinite number of cycles of stress and the accompanying cyclic strains are to a large extent plastic, it can be regarded that the notch-sensitivity of the material is small. The plastic strain tends to neutralize the stress concentration and so reduce the stress. From this view, expressed by Föppl, it follows that damping capacity is a measure of notch-sensitivity in the sense that, if the damping capacity increases, the notch-sensitivity falls off.

To the objection that certain light alloys are of small damping capacity and yet when subjected to fatigue do not start cracking at the point where the specimen has been artificially

notched, it is urged that such light alloys are extremely heterogeneous in structure, and in consequence flaws are present which reduce the fatigue strength by fifty per cent or more.

When a material is subjected to alternating stresses through undesired torsional oscillations the resistance to fatigue failure depends on the area of the hysteresis loop, i.e. on the damping capacity of the material. A material of high damping capacity, under given conditions, will be less severely stressed than one of low damping capacity. In passing, it may be mentioned that damping capacity may operate adversely since, as shown by Kimball, it may act as a cause of whirling in shafts.

Damping capacity is asserted to be a definite physical property of a material, but, as has been seen, it is a property characterized by extreme variability and one which requires super-sensitive methods for its determination.

TABLES OF PROPERTIES OF METALS AND ALLOYS

TABLE XXII

Mechanical Properties of Some Metals and Alloys

A = Annealed. C.R. = Cold rolled. D. = Drawn. H.R. = Hot rolled. C.D.H. = Cold-drawn hard.

Material	Maximum Stress in Tension lb./in.^{2}	Yield Point Elastic Limit (E.L.) Limit of Proportionality (L.P.) lb./in.^{2}	Elongation per Cent on 2 in.	Reduction of Area per Cent	Brinell Hardness Number	Impact Value C = Charpy I = Izod. ft.-lb.	Fatigue Limit (Rotating Beam Method) lb./in.^{2}	Remarks
Cast iron	31 600	—	—	—	148	—	11 000	C 3·44, Mn 0·62, Si 1·1, P 0·51, S 0·09 (H. F. Moore)
Armco iron	42 000	15 300	50	80	—	—	28 200	(N.P.L.)
Carbon steel	63 800	39 200	37	64·5	125	65(I)	26 900	C 0·1, Mn 0·72, Si 0·06, normalized (Hatfield)
	77 300	46 300	30·2	54·6	—	32(I)	29 200	C 0·3, Mn 0·67, Si 0·14, Ni 0·21, normalized (Hatfield)
	99 000	62 800	24	43·5	—	18(I)	38 100	C 0·5, Mn 0·70, Si 0·18, Ni 0·10, normalized (Hatfield)
	115 000 188 000	67 700 97 200 (L.P.)	23 10	40 29	227 380	3·3(C) 4·4(C)	56 000 97 500	C 0·93, Mn 0·38, Si 0·13, P 0·019, heat treated
Alloy steel— 3½% nickel	130 000	—	18	50	270	35(I)	—	C 0·35–0·45, Mn 0·5–0·8, Cr 0·3, hardened and tempered
Ni-Cr-Mo	224 000	—	12	25	444	15(I)	—	C 0·3, Mn 0·45, Ni 3·95, Cr 1·25, Mo 1·25, hardened and tempered
3·6% nickel	154 200	142 000	15	55	—	—	74 000	McAdam
Cr-Vanadium	201 000	164 400 (E.L.)	13·2	53·5	—	—	94 500	C 0·55, Cr 0·99, V 0·19
Si-Mn spring steel	157 000	100 000 (E.L.)	16·5	40	—	—	62 000	
Aluminium	22 600	11 300	16	65	45	—	10 500	Rolled
2. S	13 000 to 24 000	4 000 to 21 000	40–10	—	21–38	—	5 000 to 8 500	Al 99, Sp. Gr. 2·71. E = 10^6 lb./in.^{2} Compressive strength, 13 000 lb./in.^{2} (soft), 4 000 lb./in.^{2} (Y.P.) (Al. Co. of America)
No. 43-alloy	19 000	9 000	4	—	40	—	6 500	Si 5, Al 95, Sp. Gr. 2·65. Compressive strength, 25 000 lb./in.^{2}: 9 000 lb./in. (Y.P.) (Al. Co. of America)
Antimonial Lead	6 000 3 000	3 000 1 600	6 75	90 —	11·5 9·0	— —	— —	Pb 94, Sb 6, Sp. Gr. 11. Rolled or extruded. Used for tank linings, roofs, valves, and storage batteries (Scovill Mfg. Co., U.S.A.)

TABLE XXII (*contd.*)

Material	Maximum Stress in Tension lb./in.²	Yield Point Elastic Limit (E.L.) Limit of Proportionality (L.P.) lb./in.²	Elongation per Cent on 2 in.	Reduction of Area per Cent	Brinell Hardness Number	Impact Value C = Charpy I = Izod. ft.-lb.	Fatigue Limit (Rotating Beam Method) lb./in.²	Remarks
Titanium Al. bronze No. 1	65 000 to 80 000	22 000 to 26 000	25–15	24–16	90–100	—	—	Cu 90, Al 10. Sp. Gr. 7·5. Used for gears (Frontier Bronze Co., U.S.A.)
Do. No. 3	65 000 to 80 000	23 000 to 28 000	30–15	29–21	92–100	—	—	Cu 89, Al 10, Fe 1·0 (Frontier Bronze Co., U.S.A.)
Nickel silver (Omega brand)	60 000 to 125 000	—	40 1	—	—	—	—	Cu 55, Ni 18, Zn 27. Sp. Gr. 8·5, E = 20×10^6 lb./in.² Springs for electrical apparatus (Frontier Bronze Co., U.S.A.)
70/30 brass	73 200	—	20	46	—	—	17 500	Cold rolled
95/5 bronze	85 100	59 900 (L.P.)	11·7	69	166	—	27 000	Cold drawn after annealing
Manganese bronze	70 000	13 000 (L.P.)	33	41	93	—	15 000	Cu 56·8, Fe 1·5, Mn 20, Rem. Zn. E = 14×10^6 lb./in.² (R. R. Moore)
Phosphor bronze	82 200	78 800	19	70	—	—	39 900	Drawn rod
Copper	32 200	3 240 (L.P.)	56	72	47	30·5(*C*)	10 100	
Hard-drawn wire	58 000	—	—	—	—	—	—	
Ceralumin "C"	54 000	50 000 0·1% proof stress	1	—	135	—	18 500	Chill cast. Cu 2·5, Ni 1·5, Mg 0·8, Fe 1·2, Si 1·2, Cerium 0·15. Brit. Pat. No. 403700
Duralumin	64 200	40 300 36 300 (E.L.)	18·5	36·4	114	24	12 800*	Forged and heat treated. * Average of five tests by R. R. Moore
Electron	36 500	6 300 (L.P.)	17·5	20·5	64	—	17 000	E = 6×10^6 (R. R. Moore)
Magnesium alloy	35 650	18 750 0·2% proof stress	19·5 on 4 diameters	28·2	46	—	9 000	4% Al. Extruded rod.
Y-alloy	58 000	40 300	21·5	32	114	15·4	—	Forged and heat treated (High Duty Alloys, Ltd.)
Hiduminium R.R. 59	58 000 to 63 000	45 000 to 49 000 0·1% proof stress	—	—	124–128	—	—	50 000 lb./in.², 0·1% proof stress in compression 58 000 lb./in.², 0·5% proof stress in compression (High Duty Alloys, Ltd.)

TABLE XXII (*contd.*)

Material	Maximum Stress in Tension lb./in.²	Yield Point Elastic Limit (E.L.) Limit of Proportionality (L.P.) lb./in.²	Elongation per Cent on 2 in.	Reduction of Area per Cent	Brinell Hardness Number	Impact Value C = Charpy I = Izod. ft.-lb.	Fatigue Limit (Rotating Beam Method) lb./in.²	Remarks
Konel	100 000	40 000 30 000 (L.P.)	35	55	140	—	—	Forgings. Sp. Gr. 8·56, Ni 72, Fe 6·5, Co 18, Ti 2·5, Al 0·5, E=29·4 × 10⁶ lb./in.² Used for vacuum tube filaments (Westinghouse Elec. & Mfg. Co.)
70% nickel Alloy	83 500	56 000 35 000 (L.P.)	46	70	187	—	35 700*	Ni 70·27, Cu 28·64. * Direct alternating stress (N.P.L.)
Nickel . *C.R.* or *D.*	78 000 to 89 000	67 000 to 78 000	% on 4 √ area 20	—	180–200	—	35 500 *A.*	Bar. Compressive yield strength (0·1% offset), 21 300 lb./in.² *H.R.*; 56 000 lb./in.² *C.D.* Hard.
Do. *A.*	60 000 to 65 000	18 000 to 22 000	35	—	90–95	89–61(*I*)	40 500 *H.D.**	* Finished with 24% reduction, followed by stress relief anneal
H.R.	63 000 to 67 000	22 000 to 27 000	40	—	95–105	105–90(*I*)	—	Plate or bar (Henry Wiggin & Co., Ltd.)
	53 700 to 62 700	21 000 to 29 000	% on 4 √ area 15–35	—	100–125	50–80(*I*)	—	Castings
Inconel . *C.R.* or *D.*	100 000 to 127 000	89 500 to 103 000	18–15	—	200–250	100–70(*I*)	— 32 500 *A.*	Bar. Compressive yield strength (0·1% offset)
A.	78 500 to 79 500	26 900 to 31 400	35	—	120–130	20(*I*)	— 42 500 *H.D.**	Bar. * Stress relief anneal
H.R.	85 000 to 94 000	35 800 to 41 800	30	—	130–150	120–100(*I*)	—	Bar or plate
	71 600	42 500	10	—	160–170	—	—	Castings (Henry Wiggin & Co., Ltd.)

TABLE XXII (*contd.*)

Material	Maximum Stress in Tension lb./in.²	Yield Point Elastic Limit (E.L.) Limit of Proportionality (L.P.) lb./in.²	Elongation per Cent on 2 in.	Reduction of Area per Cent	Brinell Hardness Number	Impact Value C = Charpy I = Izod. ft.-lb.	Fatigue Limit (Rotating Beam Method) lb./in.²	Remarks
Monel . *C.R.* or *D.*	89 500 to 100 000	78 500 to 89 500	19	—	190–210	115–75(*I*)	—	Bar. Compressive yield strength (0·1% offset). *H.R.* 37 400 lb./in.² *C.D.* Hard, 69 400 lb./in.²
A.	67 200 to 78 500	31 400 to 38 000	35	—	110–120	120-90(*I*)	— 37 000 *A.*	Bar. * Stress relief anneal
H.R.	76 200 to 85 000	33 600 to 40 300	35	—	120–140	120–100(*I*)	— 49 300 *H.D.**	Bar and plate
	51 500 to 74 000	24 600 to 33 600	16–40	—	100–140	50–80	—	Castings (Henry Wiggin & Co., Ltd.)

TABLE XXIII

MECHANICAL PROPERTIES OF SOME STANDARD COMMERCIAL STEELS*

[Tests on bars $1\frac{1}{8}$ in. dia. at time of heat-treatment. Temperatures in ° C.
N = Normalized. *R* = Refined. *OH* = Oil-hardened. *WH* = Water-hardened. *T* = Tempered. *C* = Cementation. *Q* = Quenched.]

No.	Material	Analysis	Heat-treatment	Ultimate Stress Tons/in.²		Yield Stress Tons/in.²	Elongation on 2 in. %	Reduction of Area %	Izod Impact Test Ft.-lb.	Brinell Hardness Number 3000 kg. 10 mm. Ball		E.S.C.'s Symbol
				Min.	Max.	Min.	Min.	Min.	Min.	Min.	Max.	
1	Case-hardening Mild Steel	C 0·1–0·18, Si 0·30, Mn 0·60–0·90	*N* 900	26	35	14	28	50	—	114	153	*CHMS*
			R and *WH* *C* 900, *Q* 1st 900, 2nd 770	32	45	20	20	50	40			
2	Case-hardening 3% Ni Steel	C 0·1–0·15, Si 0·30, Mn 0·2–0·60	*N* 860	30	40	16	28	55	—	131	183	*CH3N*
		Ni 2·75–3·50, Cr 0·30	*R* and *WH* *C* 900, *Q* 1st 860, 2nd 770	45	60	30	18	45	40			
3	Case-hardening 5% Ni Steel	C 0·09–0·16, Si 0·25, Mn 0·1–0·40	*N* 850	32	42	20	28	55	60			*CH5N*
		Ni 4·6–5·50, Cr 0·30, Mo 0·50	*R* and *OH* *C* 900, *Q* 1st 830, 2nd 750	40	60	30	20	45	50			
			R and *WH* *C* 900, *Q* 1st 830, 2nd 750	60	85	50	13	40	20			
4	Case-hardening Ni, Cr Steel	C 0·09–0·13, Si 0·25, Mn 0·25–0·40	*OH* 850, *T* 600	45	55	35	20	55	45	207	255	*CHNC*
		Ni 3·0–3·75, Cr 0·80–1·10	*WH* 850, *T* 600	50	60	40	19	55	40	228	277	
			R and *OH* *C* 900, *Q* 1st 830, 2nd 770	55	70	45	14	40	22			
			R and *WH* *C* 900, *Q* 1st 830, 2nd 770	70	90	55	12	35	18			

TABLE XXIII—(*contd.*)

No.	Material	Analysis	Heat-treatment	Ultimate Stress Tons/in.² Min.	Ultimate Stress Tons/in.² Max.	Yield Stress Tons/in.² Min.	Elongation on 2 in. % Min.	Reduction of Area % Min.	Izod Impact Test Ft.-lb. Min.	Brinell Hardness Number 3000 kg. 10 mm. Ball Min.	Brinell Hardness Number 3000 kg. 10 mm. Ball Max.	E.S.C.'s Symbol
5	Case-hardening Ni, Cr, Mo Steel	C 0·14–0·18, Si 0·30, Mn 0·35–0·45 Ni 4·00–4·50, Cr 1·0–1·3, Mo 0·30	*R* and *OH*, *C* 900 *Q* 1st 830, 2nd770	85	Min.	65	12	35	25			Super *CHNC*
6	Medium Tensile Carbon Steel	C 0·25–0·35, Si 0·30,	*N* 850	26	36	16	25	50	—	114	159	Med. *C*
		Mn 0·4–0·80, Ni 0·50	*OH* 850, *T* 575	35	45	21	24	48	—	153	207	
7	High Tensile Carbon Steel	C 0·35–0·45, Si 0·12–0·30	*N* 825	35	45	18	23	40	—	153	207	*HTC*
		Mn 0·50–0·80, Ni 0·50	*OH* 825, *T* 575	40	50	24	22	45	—	183	228	
8	1% Nickel 40 Carbon Steel	C 0·35–0·45, Si 0·12–0·30	*N* 800–850	35	45	18	23	40	20	153	207	*HTCN*
		Mn 0·60–1·20, Ni 0·50–1·00	*OH* 850, *T* 500–650	40	50	—	22	45	35	183	228	
9	50 Carbon Steel	C 0·45–0·55, Si 0·15–0·30	*N* 825	40	50	20	20	35	—	183	228	*A*
		Mn 0·30–0·80, Ni 0·50	*OH* 825, *T* 575	45	60	28	18	40	—	207	277	
10	55 Carbon Steel	C 0·5–0·60, Si 0·30, Mn 0·4–0·75	*N* 830	45	—	—	18	40	—	207	—	*A31*
			OH 830, *T* 575	55	—		15	35	—	255	285	
11	Carbon Manganese Steel	C 0·10–0·15, Si 0·30, Mn 1·4–1·80	*N* 870	34	44	22	25	55	50	149	202	Supertough "A"
12	Carbon Manganese Steel	C 0·25–0·30, Si 0·30, Mn 1·4–1·80	*N* 850, *OH* 850, *T* 550–650	44	55	32	20	50	50	202	255	Supertough "B"
13	Carbon Mn, Mo Steel	C 0·15–0·20, Si 0·30, Mn 1·5–1·80 Mo 0·25–0·30, Ni 0·50–0·70	*N* 850–900, *OH* 850–900 *T* 550–650	40	50	30	20	50	70	183	228	Supertough "C20"

TABLE XXIII—(*contd.*)

No.	Material	Analysis	Heat-treatment	Ultimate Stress Tons/in.²		Yield Stress Tons/in.²	Elongation on 2 in. %	Reduction of Area %	Izod Impact Test Ft.-lb.	Brinell Hardness Number 3000 kg. 10 mm. Ball		E.S.C.'s Symbol
				Min.	Max.	Min.	Min.	Min.	Min.	Min.	Max.	
14	Low Chromium Steel	C 0·55–0·65, Si 0·20–0·50, Mn 0·50–0·80	*N* 820	55	65	30	15	—	—	255	293	*BCTB*
		Cr 0·45–0·70	*OH* 830–850, *T* 550–600	60	75	40	15	40	—	277	351	
15	3% Nickel Steel	C 0·25–0·37, Si 0·35, Mn 0·35–0·75	Normalized	35	50	20	24	45	—	152	229	*3NS*
		Ni 2·75–3·75, Cr 0·30	Oil-treated	45	60	32	22	50	35	207	277	
16	3½% Nickel Steel	C 0·35–0·45, Si 0·30, Mn 0·5–0·80, Ni 3·25–3·75, Cr 0·30	*OH* 830, *T* 530–630	55	65	45	18	50	35	255	293	*3½NS*
17	Mild Nickel Chromium Steel	C 0·18–0·25, Si 0·30, Mn 0·25–0·55, Ni 3·0–3·75, Cr 0·4–0·80	*OH* 830, *T* 500–600	50	60	40	19	55	40	228	277	*NC*
18	Super Medium Ni, Cr Steel	C 0·25–0·35, Si 0·30, Mn 0·25–0·55, Ni 3·0–3·75, Cr 0·5–0·80, Mo 0·15–0·25	*OH* 825, *T* 550–620	60	70	50	18	45	40	277	321	Super *VNCA*
19	Nickel Chromium Gear Steel	C 0·28–0·32, Si 0·30, Mn 0·25–0·55, Ni 3·0–3·75, Cr 0·55–0·70	*OH* 820, *T* 200	100	125	80	10	25	10	444	555	*VNCG*
20	Air-hardening Ni, Cr Steel	C 0·25–0·32, Si 0·30, Mn 0·35–0·60	*AH* 820, *T* 200	100	125	75	12	25	15	444	555	*SHNC*
		Ni 3·75–4·5, Cr 1·0–1·5	Harden in air, *T* 560–640	60	70	45	18	50	35	277	321	
21	Super Air-hardening Nickel Chromium Steel	C 0·25–0·32, Si 0·30, Mn 0·35–0·60	*AH* 820, *T* 200	100	125	80	12	25	15	444	555	Super *SHNC*
		Ni 3·75–4·5, Cr 1·0–1·5, Mo 0·15–0·25	Harden in air, *T* 560–640	60	75	45	17	50	35	277	351	

TABLE XXIII—(*contd.*)

No.	Material	Analysis	Heat-treatment	Ultimate Stress Tons/in.²		Yield Stress Tons/in.²	Elongation on 2 in. %	Reduction of Area %	Izod Impact Test Ft.-lb.	Brinell Hardness Number 3000 kg. 10 mm. Ball		E.S.C.'s Symbol
				Min.	Max.	Min.	Min.	Min.	Min.	Min.	Max.	
22	65-ton Nickel Cr Steel	C 0·22–0·28, Si 0·30, Mn 0·35–0·65, Ni 2·75–3·50, Cr 1·0–1·40, Mo 0·15–0·25	*OH* 830, *T* 550	65	70	57	17	40	35	293	321	*VA65*
23	Nickel Chromium Molybdenum Steel	C 0·28–0·35, Si 0·30, Mn 0·45–0·70, Ni 3·0–3·75, Cr 0·7–1·3, Mo 0·25–0·40	*OH* 830, T 550–650	60	75	50	16	40	35	277	351	Nonbrit
24	Nickel Chromium Molybdenum Steel	C 0·27–0·33, Si 0·30, Mn 0·50–0·70, Ni 2·3–2·70, Cr 0·50–0·70, Mo 0·55–0·65	*OH* 825, *T* 200–650	55	100	—	22–12	60–40	70–15	255	444	Vibrac *V30*
25	Nickel Chromium Molybdenum Steel	C 0·40–0·45, Si 0·30, Mn 0·50–0·70, Ni 2·3–2·70, Cr 0·50–0·70, Mo 0·55–0·65	*OH* 825, *T* 200–650	60	120	—	22–10	60–30	70–15	277	532	Vibrac *V45*
26	Nickel Chromium Molybdenum Steel	C 0·32–0·40, Si 0·30, Mn 0·50–0·70	*OH* 830–850, *T* 200	100	125	80	10	25	10	444	555	*VAGS*
		Ni 1·3–1·8, Cr 1·2–1·6, Mo 0·20–0·40	*OH* 830–850, *T* 550–650	50	60	50	17	40	30	277	351	
27	Case-hardening Chromium Steel	C 0·12–0·18, Si 0·30, Mn 0·40–0·60, Cr 0·9–1·20	*R* and *WH*, *C* 900 *Q* 1st 900, 2nd 770	40	60	22	18	40	25	—	—	*CHC*
28	Chromium Steel	C 0·90–1·10, Si 0·12–0·30, Mn 0·30–0·50	Annealed	—	—	—	—	—	—	—	217	*CRC*
		Ni 0·50, Cr 1·20–1·50	*OH* 820–850, *T* 150–200	—	—	—	—	—	—	600	—	
29	Copper-Molybdenum Steel	C 0·07–0·15, Si 0·15, Mn 0·60	*N* 900	24	35	14	25	50	—	111	152	*COMO*
		Copper 0·2–0·35, Mo 0·20	Annealed 900	22	32	12	28	50	—	94	140	

TABLE XXIII—(*contd.*)

No.	Material	Analysis	Heat-treatment	Ultimate Stress Tons/in.²		Yield Stress Tons/in.²	Elongation on 2 in. %	Reduction of Area %	Izod Impact Test Ft.-lb.	Brinell Hardness Number 3000 kg. 10 mm. Ball		E.S.C.'s Symbol
				Min.	Max.	Min.	Min.	Min.	Min.	Min.	Max.	
30	High Creep-resisting Steel	C 0·15–0·30, Si 0·30, Mn 0·4–0·80	*N* 875. *T* 650	32	—	18	20	45	35	137	—	*HCRS*
		Ni 0·50, Cr 0·5–1·00, Mo 0·4–0·60	*WH* 875, *T* 550–650	55	65	45	18	50	40	248	293	
31	Silico Manganese Steel	C 0·50–0·60, Si 1·50–2·00, Mn 0·60–1·00	*OH* 930, *T* 400–520	80	100	65	7	—	—	364	460	Si-Mn
32	Stainless Chromium Steel	C 0·16–0·35, Si 0·50, Mn 1·00 Ni 1·0, Cr 12·0	Air, oil, or water-treated *H* 740–940. *T* 600–750	46	66	—	20	45	30	207	300	Immaculate 2
33	Stainless Ni Cr Steel	C 0·12, Si 0·5 min., Mn 1·0, Ni 9·5–10·5, Cr 15–17	Air or water-softened from 1050° min.	35	45	12	30	55	75	150	180	Immaculate 3
34	Silicon Chromium Valve Steel	C 0·4–0·50, Si 3·25–3·75, Mn 0·40–0·60, Ni 0·50, Cr 7·5–8·5	*OH* 950–1100, *T* 700–900	55	70	40	18	45	—	255	320	Si-Cr

* *By courtesy of English Steel Corporation Ltd.*

CHARACTERISTICS AND APPLICATIONS OF THE STEELS MENTIONED IN TABLE XXIII

1. Hard surface, medium tensile strength but tough core. Used for gear wheels, cam shafts, pins, levers, spindles.

2. Medium hardness of surface and tensile strength. Used where crushing and other stresses are met with, but not severe enough to warrant the use of No. 3.

3. High tensile strength combined with very great toughness throughout. Simple in treatment. To resist high deforming stresses, e.g. gear wheels where a very strong and tough core is essential.

4. Harder surface than No. 3 with extremely high elastic limit and tensile strength combined with toughness; for heavily loaded gears, shafts, etc.

5. For exceedingly high strength of core.

6. Axles: locomotive and carriage, armature shafts, lathe and other spindles, marine shafting.

7. Crankshafts, shafts, spindles, aero-engine cylinders, front axles, connecting rods.

8. Crankshafts, spindles, aero-engine cylinders, connecting rods.

9. Gun and other mechanism parts, keys, shafts, gears, cylinders.

10. Cylinders, gears, mechanism parts, cams. Suitable for surface hardening.

11. Hard surface with medium tensile but exceptionally tough core. Used for gear wheels, cam shafts, pins, levers, spindles, railway and general engineering work.

12. Crankshafts, connecting rods and general automobile work. Tramcar and railway axles, armature spindles, railway and general engineering work.

13. Railway motion parts. Stampings of various descriptions, such as front axles for automobiles. Suitable for almost any purpose within its specified range of tensile strength, including many articles previously made in 3 per cent nickel steel.

14. Dies, mandrels, gear wheels (especially suitable for surface hardening), special casts for tyres.

15. Crankshafts, axles, connecting rods, forgings, etc.

16. Crankshafts, axles, connecting rods, forgings.

17. For highly stressed parts where carbon steels are inadequate. Bolts and studs, shafts, axles, front axles, crankshafts.

18. For use where high elastic limit, great toughness and good machining qualities are required.

Aero and engine crankshafts, propellers, shafts, gear shafts, pressure vessels.

The addition of molybdenum makes the steel very "fool-proof" in heat-treatment, so giving entirely dependable physical properties and uniformity.

19. For shock-resisting gears, etc., where great durability and strength are essential with simple treatment.

20. For highly stressed shafts, tubes, turnbuckles, gears, etc.

21. For the same purpose as No. 20. The addition of molybdenum results in a super steel which is less sensitive to unavoidable variations in heat-treatment under shop conditions. The Izod tests obtained from No. 24 are distinctly higher than those from No. 23 after similar heat-treatment.

22. For very highly stressed aero-engine connecting rods and similar duties.

23. Specially suitable for gears of heavy section, shafts of large diameter, dredger bucket pins, etc. Also for a variety of stampings used in aircraft and automobile work.

24. For the severest duties and freedom from temper-brittleness.

25. For the severest duties and freedom from temper-brittleness where high surface pressures and rubbing friction are to be resisted, such as severely loaded gear wheels.

26. When heat-treated to 100 tons per in.^{2} and over, this steel is very suitable for gears, small shafts, etc., for which a very high tensile strength is required. In the lower ranges of tensile strength it may be used with advantage in place of straight nickel chromium steel of similar alloy content.

27. Very hard surface; used for roller bearings, cams and cam shafts.

28. Ball and roller bearings, dies, mandrels.

29. For boiler and superheater tubes, boiler drums, etc., giving a much higher creep limit at 450° C. than can be obtained from ordinary carbon steels, but possessing in the cold similar physical characteristics to the latter, so that manufacturing difficulties are not increased.

30. For pressure vessels, studs and bolts for superheaters, etc., working at temperatures up to 550° C. Has a high creep limit and is remarkably free from embrittlement after long periods of service. The recommended chemical analysis within the range given will depend on the purpose for which the steel is to be used.

31. Special spring steel. Laminated and coil springs for automobiles.

32. For valve parts, pump rods, pipes and fittings, turbine blades, drop forgings and for general engineering purposes where erosion and corrosion are factors.

33. For resisting atmospheric corrosion, certain acids and sea water. Has a wide range of uses, domestic, architectural, chemical, surgical and engineering. Can be rolled into very thin sheets and drawn into tubes.

34. The exact heat treatment depends largely on composition, the higher the silicon the higher the hardening and the tempering temperatures. For use as exhaust valves for high duty and as a scale resisting steel. This steel is only suitable for high temperature work as it is decidedly notch brittle at atmospheric temperatures.

Note. For carbon steels the limiting fatigue range is from 45 to 50 per cent of the tensile strength, while for alloy steels the corresponding range is from 50 to 55 per cent of the tensile strength.

TABLE XXIV

Values of Young's Modulus and Modulus of Rigidity

Material	Young's Modulus (E) (lb./in.^{2})	Modulus of Rigidity (G) (lb./in.^{2})
Cast iron	12×10^6 to 23×10^6	5×10^6 to $7{\cdot}5 \times 10^6$
Wrought iron—		
Bar	29×10^6	$10{\cdot}5 \times 10^6$
Plate	26×10^6	14×10^6
Carbon steel	28×10^6 to 31×10^6	11×10^6 to 13×10^6
3 per cent nickel steel	28×10^6	—
30 per cent nickel steel	23×10^6	—
Nickel	$29{\cdot}5 \times 10^6$	11×10^6
Copper	15×10^6	—
Hard-drawn wire	18×10^6	7×10^6
Brass—		
Hard drawn and cast	12×10^6 to 16×10^6	5×10^6
Phosphor bronze	15×10^6 to $17{\cdot}5 \times 10^6$	6×10^6 to 7×10^6
Monel metal	26×10^6	10×10^6
Y-Alloy	10×10^6	—
70 per cent nickel alloy	26×10^6	10×10^6
Aluminium	9×10^6	$3{\cdot}8 \times 10^6$
Duralumin, heat treated	10×10^6	$3{\cdot}5 \times 10^6$

TABLE XXV

WEIGHT OF METALS

	Weight per in.^{3} (lb.)
Aluminium—	
Sheet	0·096
Cast	0·092
Aluminium bronze	0·275
Brass—	
Cast	0·301
Wire	0·307
Copper	
Cast and sheet	0·317
Wire	0·321
Duralumin	0·101
German silver	0·300
Gunmetal, 90 per cent Cu.	0·306
Iron—	
Cast	0·260
Wrought	0·280
Lead	0·410
Monel metal	0·319
Nickel	0·318
Phosphor bronze—	
Cast	0·310
Steel, average	0·282
Tin	0·262
Zinc—	
Cast	0·247
Rolled	0·260

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